

DESIGN OF HIGH SPEED, HIGH POWER SPINDLES
BASED ON ROLLER BEARINGS

By

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DESIGN OF HIGH SPEED, HIGH POWER SPINDLES
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An experimental investigation was performed on two spindle configurations based on roller bearings to determine their potential for High Speed, High Power Machining applications. The type of roller bearings considered were super precision tapered roller bearings and double row cylindrical roller bearings. The idle-load performance of each spindle was evaluated in terms of maximum operating speed, operating temperatures, lubrication requirements and required power to operate the spindle. The tapered bearing spindle was provided with a constant preloading mechanism.

Neither spindle was operated at the target 1.0 million DN (DN is the product of the bearing bore diameter in mm times the spindle speed in rpm), although both spindles exceeded the speed capabilities of current machine tools

with similar bearing arrangements. The spindle based on tapered roller bearings is strongly recommended for High Speed, High Power applications for its low power losses and low operating temperatures at 9,000 rpm, a DN value of 900,000 (the maximum speed achieved) at an operating temperature of 77 degrees Centigrade and 15 kW of power losses. The configuration based on cylindrical roller bearings is not recommended to operate above the speed of 6,000 rpm, DN value of 600,000 after which the operating temperature and power losses are above the practical limit.

CHAPTER I
HIGH SPEED, HIGH POWER MILLING

Introduction

The development of advanced cutting tools has drastically reduced the time required to perform metal removal operations. These new tool materials are capable of operating at speeds up to an order of magnitude higher than previously existing tools [1]. Thus, the use of these tools to the maximum capabilities is called High Speed Machining (HSM). These tools can be used for HSM of steel at operating speeds of 200 m/min. using Coated Carbides, or HSM of cast iron at 1000 m/min. using Silicon Nitrides or HSM of aluminum at speeds between 1000-5000 m/min. using High Speed Steels or Solid Carbides [2].

The main advantage of HSM is the capability to remove metal faster. The increased metal removal rates (MRR) is extremely attractive for such industries where machining accounts for a considerable portion of the processing time or the manufacturing cost, such as the aerospace industry, e.g. aircraft frames and engines, or the manufacturing of automotive engine blocks. Considering that by 1986 over 115 billion dollars were spent on metal removal operations [3], any increase in productivity would have substantial economic effects.

Development of High Speed Milling

The first person to investigate high speed metal removal was Dr. Carl J. Salomon in Germany from 1924 to 1931 [4]. Dr. Salomon investigated the relationship between cutting speed and cutting temperature. As a result of his investigation, Dr. Salomon concluded that as cutting speed increased, so did the cutting temperature, until a critical maximum temperature was reached. Once at this critical temperature, any further increase in the cutting speed would produce a decrease in cutting temperature. As the cutting speed was increased even further, the cutting temperature would drop to usual operating levels. Thus, around the critical temperature there is a range of very high temperatures at which tools can not operate. Below this range, usual metal removing operations were performed. Once above this critical temperature range, increased metal removal rates could be obtained if the necessary cutting speeds were achieved. The benefit of the region above the critical temperature is that the cutting speed could be increased such that infinite metal removal rates were theoretically possible. Unfortunately, Dr. Salomon's work and experimental data were mostly destroyed during World War Two, and limited information of his work is currently available.

In 1958, R. L. Vaughn, working for Lockheed, started a research program sponsored by the United States Air Force to investigate the response of some high strength materials to

high cutting speeds (152,400 surface meters per minute, smm) [4]. Some of the conclusions presented by Vaughn [4] which are of particular interest to this dissertation are

1. High speed milling could be used for machining high strength materials.
2. Productivity will increase with the use of HSM.
3. Surface finish is improved with HSM.
4. The amount of wear per unit volume of material removed decreased with HSM.
5. An aluminum alloy, 7075-T6, was machined at 36560 smm with no measurable tool wear.
6. The increase in cutting force was between 33 to 70%, over conventional machining forces.
7. At the time of investigation, the technology available could not make maximum use of HSM.

During the 1960's and 1970's various companies, such as Vought and Lockheed, experimented with HSM. In each case, the investigators agreed on the potential increase in productivity that HSM may yield [4]. Raj Aggarwal summarizes the results from published data on HSM investigations [5]. These investigations have shown that an increase in cutting speed will produce a reduction in power consumption per unit volume of metal removed (unit horsepower). Although the effect of increased cutting speed would depend on the chip load used during the investigation, in general, lower unit horsepower was obtained for higher speeds. On the effect of cutting speed on cutting forces,

results varied, while some researchers measured some decrease in the forces; others found little or no change. At this point, it is important to note that companies involved in HSM are reluctant to publish their complete results and test conditions, based on commercial competitiveness [6], which makes the comparison of results quite difficult. An area of agreement is the application of HSM to the end milling of thin aluminum ribs, where improved surface finish was obtained [5].

High Speed, High Power Machining

T. Raj Aggarwal concludes that high speed capabilities alone will not produce a relevant increase in productivity [5]. To obtain significant improvements in productivity, high speed milling must be coupled to high power machining. High power machining refers to those machining operations where the power requirements are above the capacity of common machine tools (10-20 kW). The combination of HSM and high power milling is called High Speed, High Power (HSHP) milling.

F.J. McGee [7] directed a HSM program for the Vought Corporation. As part of the research program, he identified the ideal HSHP machine tool for their investigation as having a spindle rated at 20,000 rpm speed and 75 kW power; unfortunately such a machine tool was not available. The closest available spindle was a 20,000 rpm, 22 kW spindle by Bryant. McGee [7] stresses the fact that the spindles

currently available in the market do not have the power required to make optimum use of HSM. In an effort to correct this lack of HSHP milling machines the trend has been to retrofit existing machines with high speed spindles with improved power capabilities [8]. Although this procedure will improve the HSM capabilities of the existing machine tools, there is still the need for a spindle capable of achieving spindle speeds of 10,000 rpm with power capabilities above 30 kW.

Thusty [2] defines the requirements for a HSHP spindle capable of high metal removal rates and without power limitations. For the face milling of cast iron and steel, Thusty recommends [2] the use of a spindle based on 100 mm diameter roller bearings, tapered roller bearings (TRB) or double row cylindrical roller bearings (CRB), a 10,000 rpm and 115 kW rating. Thusty shows that such a spindle could make optimum use of the new cutting tool materials. The high stiffness values which are inherent to roller bearings combined with the use of stability lobes would make the maximum use of the new cutting tools.

A spindle based on 100 mm bore TRB or CRB which operates at 10,000 rpm, would operate well above the catalogue maximum for these types of bearings. Usually these bearings are operated below 4000 rpm [9,10]. In order to operate these spindles above such speeds, special lubrication and cooling systems should be provided. The consequences of thermal differential expansions must also be

determined. If HSM is to be ever fully implemented, then HSHP spindles must be developed. For these spindles to be developed, the performance of large diameter CRB and TRB in machine tools operating at very high speeds must be researched and understood.

Goals and Scope

This dissertation is an experimental investigation on the HSHP performance of two types of large diameter (over 100 mm) bearings which are widely used in machine tools, double row cylindrical roller bearings and tapered roller bearings. The goal of this dissertation is to establish which of these bearings could best be used in HSHP spindles and what are their requirements for a successful spindle design. For each spindle configuration its lubrication and cooling requirements, its operating temperatures and its maximum operating speed must be determined. Parameters which characterize the performance of a spindle must also be defined or identified.

The design and/or development of new bearing geometries is beyond the scope of this dissertation. The high cost of developing and producing a new, non-standard bearing geometry is above the economic capabilities of the machine tool industry. However, redesigning of spindles is well within the economic bounds of the machine tool industry. The goal of this dissertation is to provide new and much needed knowledge on the HSHP performance of large diameter

bearings in machine tools and to identify those parameters which are essential for a successful HSHP spindle design based on roller bearings.

CHAPTER II

LITERATURE SEARCH

Analytical Developments

In 1963, Harris presented the first paper [11] in which an analytical method was used to predict the behavior of a bearing assembly. In this paper, Harris presents a method to estimate the operating temperature of rolling element bearings assuming steady state operation and using a finite difference scheme. The operating temperature at several different nodal points of a bearing assembly could be estimated since at each nodal point the net increase in energy is zero at steady state. By definition, at steady state, the amount of heat transferred into a nodal point equals the amount of heat transferred out of the nodal point.

According to Harris [11], the heat generated in the bearings is due to a load torque which resists the rotation of the rolling elements plus a viscous torque induced by the lubricant surrounding the rolling elements. By comparing this generated heat to the heat dissipation capacity of the assembly, the operating temperature may be estimated. Since the generated heat is the result of power losses, its computation is relevant to this dissertation.

As presented by Harris, the heat generation depends on the type of bearing used (ball or roller bearing), the bearing geometry (contact angle), loading conditions (radial or thrust), bearing diameter and lubricant properties. The load torque can be estimated using equation (2.1) [11].

$$M_x = 0.782 f_1 P_B d_m \quad (\text{N-mm}) \quad (2.1)$$

where

M_x : Load torque (N-mm).

f_1 : Load torque factor.

P_B : Equivalent applied load (N).

d_m : Bearing pitch diameter (mm).

The load torque factor is a function of bearing design and the relative bearing load. Palmgren [11] experimentally determined relations for estimating f_1 for most bearing types. For ball bearings the factor f_1 is given by

$$f_1 = z \left(\frac{P_o}{C_o} \right)^y \quad (2.2)$$

where

P_o : Static load (N).

C_o : Static Load Rating (N).

The coefficient z and the exponent y were determined experimentally and are given below in Table 2.1 from [11]. For roller bearings, the value of f_1 was also determined experimentally. The value of f_1 for several types of roller bearings is given in Table 2.2, also from [11].

Table 2.1 Coefficient z and Exponent y for Ball Bearings			
Bearing Type	Contact Angle (°)	z	y
Deep Groove	0	0.0009	0.55
Angular Contact	30	0.001	0.33
Angular Contact	40	0.0013	0.33
Thrust	90	0.0012	0.33
Self-aligning	10	0.0003	0.40

Source: Harris, T.A., "How to Predict Temperature Increases in Rolling Bearings," Product Engineering, December 1963.

Table 2.2 Load Torque Factor Values f_1 for Roller Bearings	
Bearing Type	f_1
Cylindrical	0.00025 to 0.0003
Spherical	0.0004 to 0.0005
Tapered	0.0004 to 0.0005

Source: Harris, T.A., "How to Predict Temperature Increases in Rolling Bearings," Product Engineering, December 1963.

The equivalent load P_B is a function of the type of bearing, the geometry of the bearing, and the direction of the load [11]. For ball bearings, the equivalent load is given by either equation (2.3) or equation (2.4), whichever yields the larger value of P_B . For radial roller bearings, P_B is given by equation (2.5) or equation (2.6), whichever is larger. Equation (2.7) estimates the value of P_B for thrust bearings (ball or roller).

$$P_B = 0.9 F_A \cot(\alpha) - 0.1 F_r \quad (N) \quad (2.3)$$

$$P_B = F_r \quad (N) \quad (2.4)$$

$$P_B = 0.8 F_A \cot(\alpha) \quad (N) \quad (2.5)$$

$$P_B = F_r \quad (N) \quad (2.6)$$

$$P_B = F_A \quad (N) \quad (2.7)$$

where

F_A : Axial load (N).

α : Contact angle ($^\circ$).

F_r : Radial load (N).

The lubricant flowing inside the bearing cavity will induce drag forces on the rollers. These drag forces oppose the motion of the rollers, generating heat. The expressions presented in [11] to determine the viscous torque are given as equations (2.8) and (2.9). The viscous torque is a

function of the bearing diameter, the kinematic viscosity of the lubricant, the lubrication method, and the rotational speed.

$$M_v = 9.79 \times 10^{-8} f_o (\mu \cdot n)^{2/3} d_m^3 \quad (\text{N-mm}) \quad (2.8)$$

when $\mu \cdot n > 2000$

$$M_v = 1.59 \times 10^{-5} f_o d_m^3 \quad (\text{N-mm}) \quad (2.9)$$

when $\mu \cdot n \leq 2000$

where

M_v : Viscous torque (N-mm).

f_o : Viscous torque factor for circulating oil lubrication [11]:

Angular Contact Ball Bearings (2 rows) $f_o = 8.0$

Tapered Roller Bearings $f_o = 8.0$

Cylindrical Roller Bearings (1 row) $f_o = 6.0$

μ : Kinematic viscosity (cS).

n : Rotational speed (rpm).

The heat generation rate is then the sum of the two torques M_x and M_v times the rotational speed in rpm, times a conversion factor. Thus, the heat generated at a given rotational speed under an opposing torque M is given by (2.10).

$$Q_x = 1.05 \times 10^{-4} n M \quad (\text{W}) \quad (2.10)$$

where

Q_x : Heat generation rate (W).

M : Total opposing torque $M_x + M_v$ (N-mm)

The next step in the development of bearing analysis theory was to develop an understanding of the internal behavior of the bearings. To achieve this understanding, a large research effort was undertaken during the drive to develop more efficient and reliable aircraft engines. Faster and more powerful engines required the development of more reliable bearings which could operate at higher speeds for longer periods of time. To design bearings for these operating conditions, complicated bearing analysis methods and computer programs were developed. Some of the papers which developed the understanding and modeling of high speed bearings will now be discussed in chronological order.

The next model was presented, also by Harris [12], in a paper which introduced a method to predict the occurrence of skidding in cylindrical roller bearings operating at high speed. Skidding occurs when a rolling element slides over the raceway surface instead of rolling over it. While in this condition, the cage speed is below the rotational speed of the bearing. Skidding is due to the fact that during high speed operation, centrifugal forces eliminate the normal load component acting between the rolling element and the raceway, causing the sliding of the rollers over the raceway surface. This deteriorates the roller and/or the raceway surfaces decreasing considerably the fatigue life of the bearing.

To predict skidding, it is necessary to estimate bearing internal speeds and loads. The rotational speeds of the rolling elements, cage and rollers, must be known if the effects of centrifugal force on bearing behavior are to be approximated. Since the speed of the rolling elements is affected by the loading conditions we must solve simultaneously for the loading conditions and internal speeds. The equations needed to solve for bearing internal loads and for the cage speed must first be presented.

The internal speeds of a roller are shown in Figure 2.1, from [12]. The rotational speed of the j^{th} roller is given by w_{wj} . The rotational speed of the cage is given as w_o , while the speed of the inner ring is w . The model is presented by Harris for the case when the inner race is rotating while the outer race is static. This is just the case for spindle bearing systems investigated in this dissertation. The sliding velocities can be determined as

$$V_{1j} = 0.5(d_m - D_w)(w - w_o) - 0.5D_w w_{wj} \quad (2.11)$$

$$V_{oj} = 0.5(d_m + D_w)w_o - 0.5D_w w_{wj} \quad (2.12)$$

where

V_{1j} : Sliding velocity at the inner contact of the j^{th} rolling element (m/sec).

V_{oj} : Sliding velocity at the outer contact of the j^{th} rolling element (m/sec).

d_m : Bearing pitch diameter (m).

D_w : Roller diameter (m).

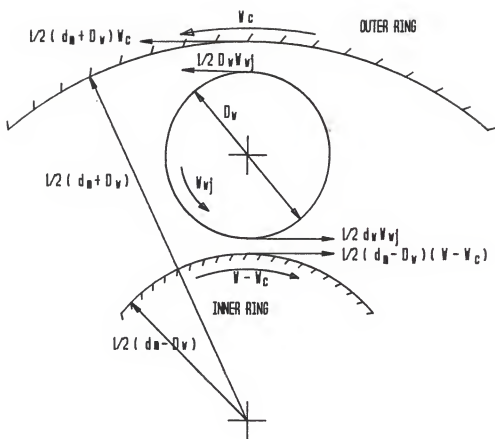


Figure 2.1 Internal Bearing Speeds, from Harris, T.A., "An Analytical Method to Predict Skidding in High Speed Roller Bearings," ASLE Transactions, July 1966.

In Figure 2.2, the loads acting on a roller are shown using the nomenclature used by Harris [12]. The i subscript refers to the inner race contact, the o subscript refers to the outer race contact, the j subscript refers to the j^{th} roller, while y and z subscripts indicate horizontal and vertical components respectively. Thus, the load Q_{zoj} , indicates a vertical load, acting on the outer race contact of the j^{th} roller. The loads Q_{zoj} and Q_{zioj} are the reactions to external applied loads acting on the j^{th} roller. Load F_{aj} is caused by the cage acting on the roller. Loads Q_{yoj} and Q_{yioj} are loads caused by the fluid pressure acting on the rollers at each roller raceway contact, while the drag forces acting at each contact are given by F_{oj} and F_{ioj} . The boldface version of the previous are the dimensionless forms of the corresponding loads. The effect of high speed operation on the roller, which induces a centrifugally oriented force, is F_w . The elastohydrodynamic loads are introduced by Harris here.

During steady state operation, the summation of the forces acting on each roller, in directions y and z , must equal zero. In dimensionless terms the force balance is given by [12] as

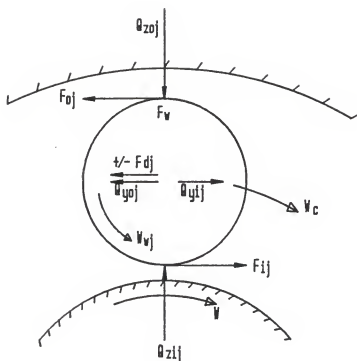


Figure 2.2 Loads Acting on a High Speed Roller
 Harris, T.A., "An Analytical Method to Predict
 Skidding in High Speed Roller Bearings," ASLE
 Transactions, July 1966.

$$\frac{R_o}{R_1} (Q_{z1j} + F_w) - Q_{z0j} = 0 \quad (2.13)$$

$$Q_{y1j} + F_{1j} - \frac{R_o}{R_1} (Q_{y0j} - F_{0j} \pm F_{a1j}) = 0 \quad (2.14)$$

where

R_o : Equivalent external radius of the cylinder
(mm).

R_1 : Equivalent internal radius of the cylinder
(mm).

$$Q_{z1j} = \frac{Q_{z1j}}{l_w E' R_1} \quad (2.15)$$

$$Q_{z0j} = \frac{Q_{z0j}}{l_w E' R_o} \quad (2.16)$$

$$Q_{y1j} = \frac{Q_{y1j}}{l_w E' R_1} \quad (2.17)$$

$$Q_{y0j} = \frac{Q_{y0j}}{l_w E' R_o} \quad (2.18)$$

$$F_w = \frac{F_w}{l_w E' R_o} \quad (2.19)$$

$$F_{a1j} = \frac{F_{a1j}}{l_w E' R_o} \quad (2.20)$$

l_w : Roller length (mm).

$$E' = \frac{E}{1 - \sigma^2} \quad (2.21)$$

E: Modulus of elasticity (N/m²).

σ : Poisson's Ratio.

The lubricant induced loads F_{1j} , F_{0j} , Q_{y0j} and Q_{y1j} are given next, in dimensionless form, as presented by Harris in [12].

$$F_{1j} = -9.2G^{-0.3} U_{1j}^{0.7} + \frac{V_{1j}I_{1j}}{H_{1j}} \quad (2.22)$$

$$F_{0j} = -9.2G^{-0.3} U_{0j}^{0.7} + \frac{V_{0j}I_{0j}}{H_{0j}} \quad (2.23)$$

$$Q_{y0j} = 18.4 (1-\tau)G^{0.3} U_{1j}^{0.7} \quad (2.24)$$

$$Q_{y1j} = 18.4 (1-\tau)G^{0.3} U_{1j}^{0.7} \quad (2.25)$$

where

$$G = \alpha E'$$

α : Is the pressure coefficient of viscosity (mm²/N).

$$H_{1j} = 1.6 \frac{G^{0.6} U_{1j}^{0.7}}{Q_{x1j}^{0.13}} \quad (2.26)$$

$$H_{0j} = 1.6 \frac{G^{0.6} U_{0j}^{0.7}}{Q_{x0j}^{0.13}} \quad (2.27)$$

$$V_{1j} = \frac{\mu_o V_{1j}}{E' R_1} \quad (2.28)$$

$$V_{o,j} = \frac{\mu_o V_{o,j}}{E' R_o} \quad (2.29)$$

$$U_{1,j} = \frac{\mu_o U_{1,j}}{2E' R_1} \quad (2.30)$$

$$U_{o,j} = \frac{\mu_o U_{o,j}}{2E' R_o} \quad (2.31)$$

$$I_{1,j} = 2 \int_0^{4q_{1,j}} e^{Gq_{1,j}[1-(y/4q_{1,j})^2]^{1/2}} dy \quad (2.32)$$

$$I_{o,j} = 2 \int_0^{4q_{o,j}} e^{Gq_{o,j}[1-(y/4q_{o,j})^2]^{1/2}} dy \quad (2.33)$$

As it can be seen from above, the elastohydrodynamic loads are non-linear functions of the roller speeds and lubricant properties. Note that the operating temperature is an input to the analysis and it is not corrected for each iteration.

The model provides a method to solve for the cage speed w_o . If there is skidding, the cage speed will be below the expected value of

$$w_o = 1/2 w(1 - D_w/d_m) \quad (2.34)$$

which is the cage speed during rolling motion.

To determine the cage speed equations (2.13) and (2.14) are not enough. Torque balances must be performed at each bearing location and for the complete bearing. This would provide the necessary equations to solve for cage speed w_o and roller rotational speed $w_{o,j}$, cage load on the roller $F_{a,j}$,

and outer race contact load $Q_{\pi o j}$. As it was noted before, the other loads are non-linear functions of lubricant properties and roller speed. The inner race contact loads $Q_{\pi i j}$, are computed from static load analysis of the complete bearing.

The solution method would require the computation of an initial cage speed from a known inner race speed. Using this cage speed, the conditions at each roller location are then computed. The loads are added up and they must balance the externally applied loads. Harris does not solve the model in this manner, due to the required computational tools which were not available to him at the time. Instead, he introduces some simplifying assumptions:

1. Since not all the rollers are loaded, Harris only considers the loaded contacts.
2. At steady state, the speed and load conditions at any loaded roller location is the same as in the most heavily loaded roller. The drag force acting on a loaded roller is determined by dividing the computed drag force by the number of loaded rollers.

These simplifications drastically reduce the number of computations needed to solve for the roller speeds. Still, Harris's analysis yields a sufficiently close prediction of the occurrence of skidding. It shows that skidding does not exist in preloaded bearings. As soon as the centrifugal effects remove the preload, skidding starts. In those

applications where out of roundness bearings are used, the geometry of the bearing improves skidding behavior.

The simplifications do take a toll of the accuracy of the model. The simplified model can predict the occurrence of skidding but would not quantify it. The model is also limited for heavily loaded bearings. With the development of the digital computer, the complete analysis developed by Harris will later be used by other researchers in the development of more accurate models, as it will be shown below.

Boness [13] provides some experimental data which corroborates the results obtained by Harris in his simplified model. At the same time, the experimental results aroused some doubt on the validity of Harris's simplifying assumptions. The results presented in [13], show that for each roller, the rotational speed is different. The oil film thickness is also different at each roller location. This explains the limitations of Harris's model. Boness also found that by decreasing the amount of lubricant in the bearing cavity, the amount of skidding could be reduced by 75 percent. To obtain this amount of reduction in skidding, a very small amount of oil must be used; which is not always possible since at high speed applications oil provides the only reliable source of cooling.

Poplawski [14] presents an analytical model which is based on the model developed by Harris [12] and the experimental results presented by Boness [13]. In his model, Poplawski considers the rotational speed of each roller as an independent variable which must be solved for in order to compute the operational conditions of the roller. Poplawski's model is quite similar to the complete analysis presented by Harris [12], but no simplifications are necessary thanks to the availability of powerful computers. It also includes the computation of the drag forces at each roller location.

In Figure 2.3 the loads acting on a high speed roller, are shown according to Poplawski [14]. The similarity between this model [14] and the one presented by Harris [12] is obvious. In the model shown in Figure 2.3, there is an extra load acting on the roller, which is a drag force caused by the cage driving the roller and it is labeled $f_p F_{d3}$. Therefore, rewriting equations (2.13) and (2.14) to include this term equations (2.35) and (2.36) are obtained.

$$\frac{R_o}{R_1} (Q_{x13} + F_w \pm f_p F_{d3}) - Q_{x03} = 0 \quad (2.35)$$

$$Q_{y13} + F_{13} - \frac{R_o}{R_1} (Q_{y03} - F_{03} \pm F_{d3}) = 0 \quad (2.36)$$

Another difference between this model and the original is the computation of the deflection of the rollers. Harris [12] uses a load deflection behavior which ignores the

lubricant film between the rollers and the raceways. Poplawski's model [14], does includes the deformation of the oil film between the rollers and the raceways. He modifies the deflection equation to

$$\delta_j = \delta_x \sin \phi_j + \delta_y \cos \phi_j - (G_1/2) + h_{1j} + h_{oj} \quad (2.37)$$

$$\delta_j = \delta_{oj} + \delta_{1j} \quad (2.38)$$

where

$$\delta_{1j} = \left(\frac{P_{1j}}{K'} \right)^{0.9} + P_{1j} \left(\frac{\delta h_{1j}}{\delta h_{1j}} \right) P_{1j} \quad (2.39)$$

$$\delta_{oj} = \left(\frac{P_{oj}}{K'} \right)^{0.9} + P_{oj} \left(\frac{\delta h_{oj}}{\delta h_{oj}} \right) P_{oj} \quad (2.40)$$

where

h: Oil film thickness given by

$$h = \frac{8}{3} \frac{\alpha^{0.6} (\mu_o u)^{0.7} E^{1.0.3} R^{0.13} l_w^{0.13}}{p^{0.13}} \quad (2.41)$$

$$\frac{\delta h}{\delta p} = \frac{-0.302 \alpha^{0.6} (\mu_o u)^{0.7} E^{1.0.3} R^{0.13} l_w^{0.13}}{p^{1.13}} \quad (2.42)$$

The deflection behavior was used to determine the inner race contact loads as

$$F_x = \sum_{j=1}^n P_{1j} \sin \phi_j \quad (2.43)$$

$$F_y = \sum_{j=1}^n P_{1j} \cos \phi_j \quad (2.44)$$

which is the same method used by Harris.

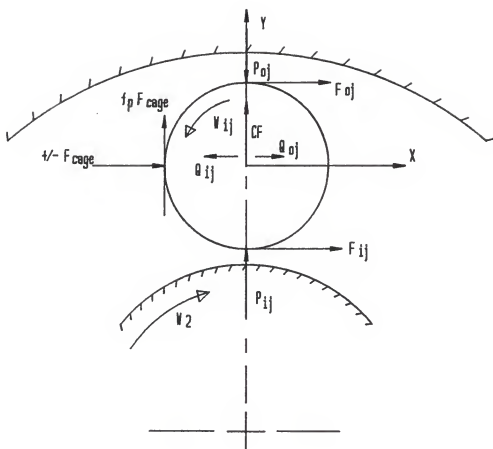


Figure 2.3 Loads acting on a High Speed Roller from Poplawski, J.V., "Slip and Cage Forces in a High Speed Roller Bearing," ASME Journal of Lubrication Technology, April 1972.

One major development of Poplawski's model is the evaluation of the drag forces in more detail than in previous models. The equations used to determine the drag forces are as presented by Harris [12] but now evaluated at each roller location. The drag force acting on an unloaded roller with translatory motion is given [14] as

$$\begin{aligned} F_{drag} &= F_{0j} - F_{1j} \quad (N) \\ &= 9.2(1+2\tau)G^{-0.3}U_{ou}^{0.7} \quad (2.45) \end{aligned}$$

There is also considerable friction between the cage and the guiding surfaces, either in the outer race or the inner race, depending on which side is used for cage riding. For inner race rotation and inner race guiding, Poplawski suggests that the force is given by

$$F_{pilot} = f_p N \quad (N) \quad (2.46)$$

where N is the normal force acting on the pilot.

The last drag force component to be considered is due to the churning of the oil by the rollers and Poplawski introduces the following relationship

$$F_{churn} = 1/2 \rho C_D S_e V_o^2 \quad (2.47)$$

where

ρ : Effective density of the mixture = oil

C_D : Drag Coefficient

S_e : Effective Drag Area (mm^2)

V_o : cage orbital velocity (m/sec.)

As before, the force balance equations are not enough to solve for the unknown variables, namely F_d , w_c , w_{wj} and P_{oj} . Torque balances are performed, based on an initial cage speed, for each loaded roller to estimate the rotational roller speed. Once the speed is computed for all rollers, the drag forces acting on the unloaded rollers and on the cage are estimated. Then a second torque balance is performed for the complete bearing assembly between the drag forces and the cage loads. If equilibrium does not exist, the cage speed is corrected and the roller conditions are computed once again. This procedure is iteratively repeated until equilibrium of the complete bearing is achieved.

Poplawski's model has very good correlation with the experimental data presented by Boness [13]. It is a more complete model in the fact that it includes the speed of each roller as an independent variable. The incorporation of the individual roller drag forces makes of it a more realistic model. The work presented in [14] helped in the further development of the bearing analysis methods.

In an effort to quantify the heat generation rates, Witte [15] derived some theoretical equations, which were later modified to accommodate experimental results for tapered roller bearings. A heat generation potential factor G was developed based on the geometry of a tapered roller bearing under pure thrust load. The author called this factor G ; it is based strictly on the geometry of the bearing and it is a constant for a particular bearing

series. The G factor can be obtained from a tapered bearing catalog or computed according to the equation given by Witte in [15] as

$$G = \frac{D^{1.5}}{D^{-0.17}(nl)^{-2/3} (\sin \alpha)^{1/3}} \quad (2.48)$$

The G factor is related to the resisting torque of the bearing. The lower the G factor is, the lower the heat generation for that bearing. The relationship between pure axial load, the G factor, and resisting torque is given by Witte [15] as

$$M = 1.1 \times 10^{-4} G F_a (S\mu\phi)^{0.5} (F_a\phi)^{1/3} \quad (2.49)$$

where

M: Resisting torque (lb.-in.).

G: Bearing Geometry Factor.

F_a : Axial Load (lb.).

S: Bearing Speed (rpm).

μ : Lubricant viscosity at atmospheric pressure (cP).

ϕ : Lubricant pressure-viscosity index (in²/lb).

and it is limited for (S μ) values larger than 3000 and for axial loads which are less than twice the axial load rating of the bearing.

For the case when radial loads are applied instead of an axial load, equation (2.49) should be modified to compensate for the different orientation of the load. Equation (2.50) gives the relation between M, G, and radial loads.

$$M = 1.1 \times 10^{-4} G (S\mu\phi)^{0.5} (f_T F_T / K)^{1/3} \quad (2.50)$$

where

f_T : Equivalent thrust load factor

K: Ratio of basic dynamic radial load rating to
basic dynamic thrust load rating.

and it is limited for $(S\mu)$ values larger than 3000 and for radial loads which are less than twice the radial load rating of the bearing. The f_T and K factors can be obtained either from [15] or from the manufacturer of the tapered bearing.

Witte obtained good correlation between his equations and experimental data. One shortcoming of his experiments was that he used less than 1.9 liters per minute of lubricating oil. This is quite low compared with what is commonly used in high speed bearing applications.

Astridge and Smith [16] performed an experimental investigation in an attempt to quantify the power losses, and heat generation in high speed cylindrical roller bearings. They used bearings with bore diameters of 300 and 311 mm, operating them at 1.1 million DN. The bearings were operated with diametral clearance, simulating operating conditions in aircraft engine bearing applications. From

their experimental results and other published data, Astridge and Smith [16] suggested 10 sources of heat generation:

- 1) Viscous dissipation between rollers and races.
- 2) Viscous dissipation between rollers ends and guide lips.
- 3) Elastic hysteresis in rollers and races.
- 4) Dissipation in films separating cylindrical end faces of rollers and cage.
- 5) Dissipation in films separating cage and traces.
- 6) Dissipation in films separating cage side faces and chambers wall.
- 7) Displacement of oil by rollers.
- 8) Flinging of oil from rotating surfaces.
- 9) Oil feed jet kinetic energy loss.
- 10) Abrasive wear and asperity removal.

As it can be seen from the list, most of the sources are due to the drag forces acting between the rollers and the lubricant. The lubricant is displaced by the rollers as they move within the bearing cavity. According to Astridge and Smith [16] the single most important source of heat generation is due to the churning of the oil between the rollers and the raceways. In the case considered in [16] not all the rollers were loaded.

Performing a parametric study, Astridge and Smith identified which of the parameters related to bearing operation have a larger influence on heat generation. The ones with a stronger effect on heat generation were found to be speed, oil flow rate, oil viscosity and pitch diameter.

In [17] Rumbarger et al., presented a sophisticated computer analysis for single row high speed cylindrical roller bearings. The authors incorporate into a single model the load-deflection behavior, the kinematic and the EHD behavior and the thermal behavior. Previous models did not consider the effects on bearing behavior of the interaction between these components of bearing performance. In the model presented by Harris in [12] a single overall bearing temperature is considered, while in [17] the temperature at each contact is computed based on the kinematics, the EHD conditions and the loads present at that contact. In contrast with Poplawski's model, in the model presented in [17] the drag forces are computed for each roller element using the estimated EHD conditions for the speed and temperature estimated for each contact. These loads are then compared with previous speed and temperature iterations, which if different are corrected. If the kinematic conditions in a roller location change, the overall load distribution may be affected, causing a change in the elastohydrodynamic conditions and in thermal performance. Due to the iterative solution method used, the computation needs are enormous. It is then necessary to

limit the analysis to steady state operation, otherwise the required computation capabilities would make the codes too complex and expensive to use.

In this model, the elastic, kinematic, and thermal analysis are similar to the ones used in the models presented by Harris and Poplawski which were discussed previously. The model presented by Rumbarger et al. in [17] is relevant since it introduces the use of a complete fluid analysis to evaluate the viscous effects of the lubricant on the rolling elements. Therefore, the discussion of the model would be concentrated into this new development presented in [17].

The authors in [17] identified two main viscous drag torque sources. The first source is the viscous drag caused by the rolling elements moving through the lubricant. As the rollers rotate within the bearing cavity, the lubricant flows around them and between the rollers and the guiding surfaces. The second source of viscous drags according to Rumbarger et al. is caused by the motion of the cage within the bearing. As the cage rotates, it is in contact with the lubricant at the inner and outer surfaces, at the lands and at the side surface. The total drag torque is the sum of the drag torques acting on each rolling element plus the drag torque acting on the cage.

The total drag torque acting on a roller is the sum of the drag torque acting on the roller surface, plus the drag torque acting on the roller end, plus the retarding torque

caused by the contact between the roller and the cage. The torque acting on the roller surface is computed by the authors of [17] as

$$T = \tau_w A r \text{ (N-m)} \quad (2.51)$$

where

T : Drag torque acting over the element surface
(N-m).

τ_w : Wall Shear Stress (N/m²).

A : Surface area of the roller (m²).

r : Reference radius from the center of rotation
(m).

The authors [17] recommend for the computation of the shear stress acting on the rollers equation (2.52).

$$\tau_w = f(1/2 \rho U^2) \text{ (N/m}^2\text{)} \quad (2.52)$$

where

f : Friction factor computed from the Reynolds
number assuming turbulent flow [17].

ρ : Fluid mass density (oil and air mixture)
(Kg/m³)

U : Mass average velocity of the fluid (m/sec)

To compute the drag torque acting on the roller ends the authors recommend equation (2.53).

$$T_{end} = 0.5 w^2 r^5 C_n \text{ (N-m)} \quad (2.53)$$

where

T_{end} : Drag torque acting on the end of the roller
(N-m)

w : Rotational speed of the roller (rad/sec.)

C_n : Correlation factor:

$3.87/(N_{Re})^{0.5}$ for laminar flow

$N_{Re} < 300,000$

$0.15/(N_{Re})^{0.17}$ for turbulent flow

$N_{Re} > 300,000$

The last torque component acting on the rollers is due to the contact between the rollers and the cage. To estimate this torque the authors recommend the following equation

$$T_{add} = \mu_c \frac{F_{jg}}{A_g} \left[\sum_{N=1}^N \frac{|V_{R1} - V_{H1j}|}{V_{R1} - V_{H1j}} A_{H1j} r_{H1j} \right. \\ \left. - 2 \sum_{k=1}^{S/2} A_{Vjk} r_{Vjk} \right] \quad (2.54)$$

where

F_{jg} : Contact force between the roller and the guiding surface (N).

N : Number of horizontal lamina.

V_{R1} : Velocity of the race at the i^{th} horizontal lamina (m/sec.).

V_{H1j} : Velocity of the roller at the i^{th} horizontal lamina (m/sec.).

S : Number of vertical lamina.

T_{add} : Torque produced at the roller-cage contact (N-m).

μ_c : Friction coefficient between the roller and the guiding shoulder.

A_{Hj1} : Area of the i^{th} horizontal lamina (m^2).

r_{Hj1} : Distance from the i^{th} horizontal lamina to the center of the roller (m).

A_{Vjk} : Area of the k^{th} vertical lamina (m^2).

r_{Vjk} : Distance from the k^{th} vertical lamina to the center of the roller (m).

A_G : Total contact area between the rollers and the guiding surfaces (m^2).

which is obtained by dividing the contact area into various vertical and horizontal lamina.

To compute the torque induced by the cage moving through the lubricant, equation (2.51) is used for the inner and outer surfaces of the cage and for the lands. For the sides, equation (2.53) is used.

The main problem of the fluid model is its sensitivity to the amount of oil inside the bearing cavity. The authors used a volume percent of 15 to 20%; the percent of the total bearing cavity volume which the oil occupies. These values of volume percent yielded good correlation between the experimental results and the model computations. The accuracy of this procedure is questionable, since there is no reliable way to measure the amount of oil inside the bearing cavity. The density used for the mixture is computed based on an amount of oil present in the cavity, which is difficult to determine. The major contribution of the model presented in [17] is the use of an

interdisciplinary approach to solve for the operating conditions of a high speed bearing [18].

Since the model in [17] was presented, several advanced computer codes have been developed for the analysis and design of high speed bearings. The driving force for the development of these codes have been the need for more reliable bearings for combat aircraft mainshaft bearings [19]. Two main types of bearing analysis codes have been developed, for quasistatic or steady state analysis and for dynamic or transient analysis. The first is represented by programs such as SHABERTH for the analysis of shaft-bearing systems, and CYBEAN, for the analysis of cylindrical roller bearings [20]. The dynamic analysis codes are represented by the program DREB, which is used to analyze the transient behavior of ball and roller bearings [20].

A major shortcoming of these computer codes is that their results are seldom compared to experimental results as pointed out by Parker in [19]. Another problem pointed out by Parker [19], is that even if comparison to experimental results is intended, there are some computations which cannot be compared since there is no experimental way to obtain experimental data to match the computations. For example, some programs include in their output roller skew angles and element temperatures which are yet to be measured experimentally. Another problem with the computer codes is the dependency on the volume percent of oil in the bearing cavity to estimate the thermal behavior

of the assembly. Those researchers which have attempted a comparison between the computer results and experimental data are required to chose such a volume percent such that their computations approximate the experimental results [17, 19,21,22,23].

Although the computer programs are still to be improved, they have facilitated the development of advanced bearing designs. The use of an interdisciplinary approach to the analysis of the behavior of high speed bearing can only be done using the computer. The problem is too complex to be solved by a single individual without the assistance of a high speed computer. The codes currently can only be used in high speed supercomputers, which means there are not available to most engineers involved in designs with bearing applications.

High Speed Bearings: Experimental Results

In 1974, Signer et al. [24] presented experimental data on high speed angular contact ball bearings. ACBB of 120 mm diameter, 20° and 24° contact angles were tested to 3 million DN. The test conditions were made to simulate the operating conditions in an aircraft turbine.

It was found in this investigation [24] that power losses increased linearly with speed and with increased oil flow rate through the inner race. Inner race lubrication was more effective than other lubrication in reducing the

operating temperature, for the same oil flow rate. It was interesting to find that when the oil flow rate was increased over 3.8×10^{-3} cubic meters per minute (1.0 gpm), the temperature increased, probably due to the increased quantity of lubricant within the bearing cavity and to the resultant churning.

Parker and Signer [25] present the results of their investigation of high speed tapered bearings. The bearings used had 120.65 mm bore with capability to use either jet lubrication or cone-rib lubrication. The use of cone-rib lubrication proved to be more efficient in limiting the operating temperature. It was also found that the use of cone-rib lubrication instead of jet lubrication reduced the power consumption.

The experiment showed that the bearing temperatures and power losses increased with spindle speed. The effect of load on bearing temperature was insignificant.

In [25], Parker and Signer presented results of their testing of TRB to DN values higher than one million. Since TRBs have a better load-deflection characteristic than ACBBs or CRBs for the same envelope, they are preferred for some applications where weight or space are critical. It was also demonstrated that by providing the cone-rib/roller end contact with sufficient lubrication, TRB can be operated to very high speeds. The lubrication method recommended then was the use of holes drilled through the cone, through which oil was forced into the cone-rib area.

Parker and Signer used specially designed TRB to investigate the high speed performance with cone-rib lubrication versus the performance with oil jet lubrication. The bearings used were of standard design but provided with cone-rib lubrication to improve their high speed operating performance.

The bearing tested had a bore diameter of 120.6 mm, an outside diameter of 206.4 mm, a cup angle of 34° , and it contained 25 rollers. The test speeds were 6,000, 10,000, 12,500 and 15,000 rpm. The oil flow rates used were 1.9×10^{-3} to $15.1 \times 10^{-3} \text{ m}^3/\text{min}$.

The test results obtained in [25], showed that cone-rib lubrication plus jet oil lubrication was a better lubrication arrangement than oil jet lubrication alone. In fact, the higher speeds could not be achieved safely with oil jet lubrication alone. As for oil flow rate, by increasing the oil flow rate, temperatures decreased while power losses increased. It was also shown that for oil flow rates over $11.4 \times 10^{-3} \text{ m}^3/\text{min}$, a further increase in oil flow will not produce a significant temperature decrease. Observing the power losses induced by the increased oil flow rate, the use of oil flow rates larger than $11.4 \times 10^{-3} \text{ m}^3/\text{min}$ do not seem justifiable.

Spindle speed also produced considerable increase in temperature and power losses. The effects of load on bearing temperature were insignificant compared to the effects of the oil flow rate and spindle speeds tested. The

authors of [25] used the equation derived in [15] to estimate the heat generation rates. The power losses estimated using the equation from [15], had good correlation with the experimental results.

Parker et al. [26] presented results of computer optimized TRB bearings. These bearings were designed by first optimizing the standard TRB design, as the ones used in [25]. The optimized design was then presented to a leading TRB manufacturer who suggested changes which would allow the bearing to be economically manufactured. The bearings used in [26] used 23 rollers, it had a cup angle of 31° , 120.65 mm bore diameter and outer diameter of 190.5 mm.

The bearings were provided with cone-rib lubrication and instead of oil jet lubrication, the front of the bearing was lubricated through holes in the cone and through the spindle. Oil was forced centrifugally through these holes into the front of the bearing. Test speeds varied from 6,000 rpm to 20,000 rpm. Oil flow rates varied from 3.8×10^{-3} to $15.1 \times 10^{-3} \text{ m}^3/\text{min}$.

The computer optimized bearing operated at lower temperatures, lower power losses and higher spindle speeds than the standard bearing. Effects of oil flow rate, spindle speed and load on bearing temperatures and losses were similar for the optimized bearing and the standard bearing.

Currently, aircraft engines operate at a maximum DN value of 2.4 million [23,27]. The mean time between bearing removal is up to 3000 hours from 300 hours ten years ago [34]. Improvements in the lubrication methods have allowed researchers to operate ACBB and CRB to 3.0 million DN, while TRB have been operated to a 2.4 million DN [23,27]. The use of AISI M-50, a vacuum-induction melted, vacuum arc melted alloy, has greatly improved the fatigue life of high speed bearings.

CHAPTER III

EXPERIMENTAL EQUIPMENT

High Speed, High Power Milling Machine

The Machine Tool Laboratory at the University of Florida is equipped with a HSHP milling machine, shown in Figure 3.1. The spindle is driven by a 115 kW, 3000 rpm, ASEA D.C. motor by means of a two stage flat belt transmission. The first stage is a belt from the motor to the intermediate shaft, located in the column of the milling machine. The second stage, is from the intermediate shaft to the the spindle. The speed ratio used for the high speed test between the motor and the spindle was 0.26.

The spindle is mounted on the HSHP milling machine on the front, bolted to a mounting bracket. Lubrication connections and instrumentation are external to the HSHP machine, making the change of spindles a simple task. To change the spindle mounted on the machine, the current spindle is unbolted and removed using a hoist. The next spindle can then be mounted and bolted. The lubrication system can easily be modified to accommodate several spindle designs.

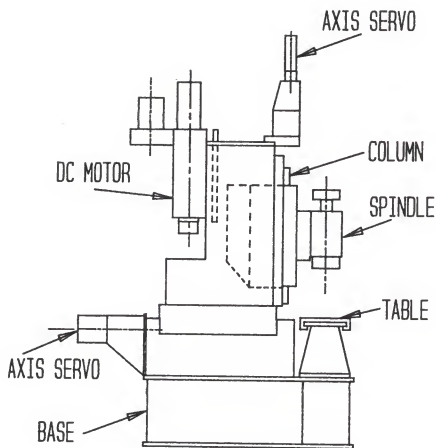


Figure 3.1 HSHP Milling Machine

This HSHP milling machine permits a complete investigation of the configurations under study. Each configuration is tested not only for idle operation performance, but also for cutting capabilities and chatter stability.

Test Spindles

The two spindle bearing configurations shown in Figure 3.2 and Figure 3.3 were tested for HSHP performance. Their operating temperatures, lubrication needs and power demands were investigated at several speeds, during idle, no load operation. Both spindles were equipped with circulating oil lubrication. The amount of oil circulated was varied from 1.5 liters to 3.8 liters per minute, per bearing. The spindles were tested for maximum operation speed.

Configuration I is based on double row cylindrical roller bearings (CRB) NN 30K/SP manufactured by SKF. It has one NN3019K/SP on the drive side and a NN3022K/SP on the tool side. The CRBs support the radial loads while the thrust load is supported entirely by a Radiax, a 234420 BMI/SP series angular contact thrust ball bearing (ACTBB) by SKF, with a contact angle of 60°. This configuration is sometimes referred to as Standard Configuration I by SKF researchers [28].

The preload in this configuration is provided, individually for each bearing. The radial bearings are preloaded radially by eliminating any clearance between the

outer race and the rollers. As it can be seen from Figure 3.2, tightening the nut A pushes on the inner race of the lower CRB, moving the inner race and the rolling elements up the tapered. As the rolling elements are driven up the taper of the spindle, the diametral clearance between the elements and the outer race is reduced. Tightening the nut further, contact between all the rollers and the outer race is produced, completely eliminating any clearance. If nut A is tighten even more then interference is produced.

The ACTBB is preloaded by tightening the nut B to press together the bearing assembly. As the nut B is tighten, any gap between the races and the bearing spacer C is eliminated. Once the nut B is completely tighten, the preload between raceways and balls is achieved. The preloading of this configuration is done during the assembly of the spindle and cannot be released, unless the spindle is completely disassembled.

The maximum speed achieved by this configuration was 8,000 rpm. The operation temperatures were above the recommended for the type of oil used. The power losses were almost 14 kW, which means that for a 20 kW milling machine could only perform 5 kW of useful work at 8,000 rpm. When the spindle was driven over 8,000 rpm, the ACTBB failed within seconds of starting the test. This happened twice: at 9,000 and 10,000 rpm. The failure was too fast for the PROMESS sensor to detect any increase in the load of the bearings. After discussing the failure with SKF

researchers, it was concluded that the cause of the failure was the loss of preload. The loss of preload induced skidding, which was the mode of failure of the bearing. To correct the problem, the mounting preload must be increased and a larger amount of lubricant must be provided to the upper raceway. To achieve this increase in preload, the spacer separating the two raceways, spacer C, must be ground, bringing the two raceways closer together. This increase in preload would also induce an increase in bearing temperature, which could not be permitted, since operating temperatures are already too high.

Configuration II is based on TRB. This configuration operates under constant preload. A constant preload is maintained by the bearing in the drive side, the HYDRA-RIB_{TM} by TIMKEN, Figure 3.4. The bearing is provided with a hydraulic chamber and piston mechanism which provide a load to the back of the rollers. As the chamber is pressurized, the piston displaces forward, pushing on the rollers. This forward displacement of the rollers produces the diametral interference or preload. The preload force is proportional to the hydraulic pressure in the chamber. If during the operation of the spindle the loads acting on the rollers increase, the piston would retract to a point where the load on the rollers equals the preset value. If on the other hand, the load on the bearing is reduced during the operation of the spindle, the piston would move forward until the preset load on the rollers is re-established.

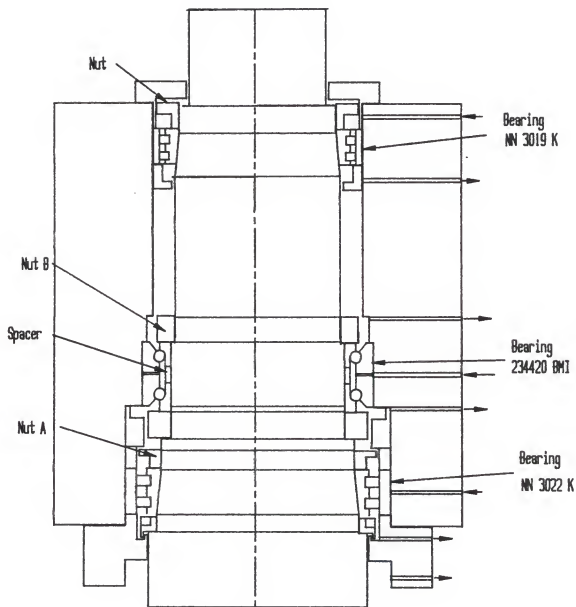


Figure 3.2 Configuration I Test Spindle

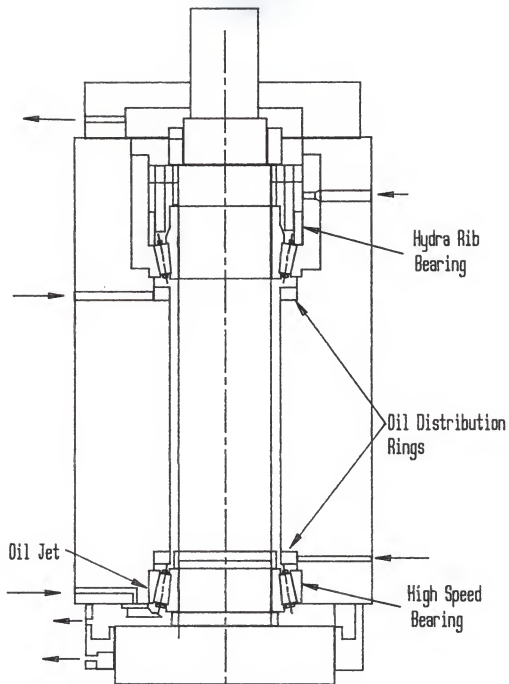


Figure 3.3 Configuration II Test Spindle

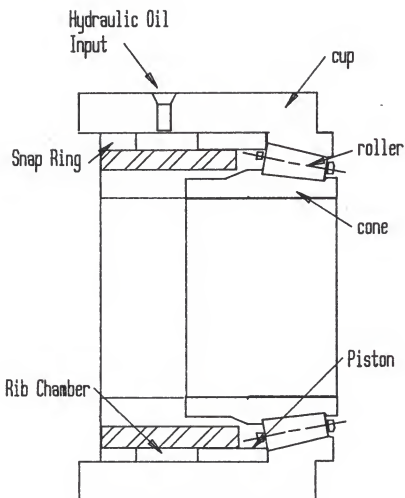


Figure 3.4 HYDRA-RIB™ Bearing

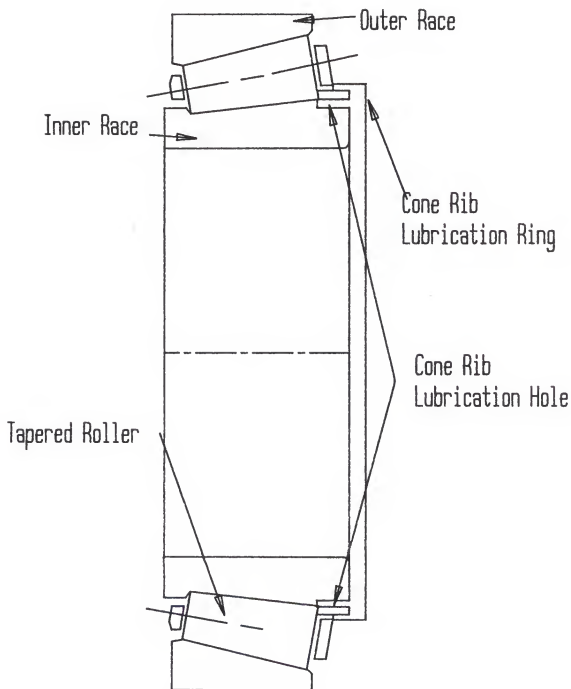


Figure 3.5 High Speed Bearing with Cone-Rib Lubrication

The High Speed (HS) bearing, 100 mm diameter, in the tool side, Figure 3.5, is provided with cone-rib lubrication. The cone is lubricated through holes drilled from the back of the bearing to the cone-rib. At the back of the bearing, there is a ring which entraps the oil supplied by jets forcing it centrifugally into the holes.

This configuration operated successfully up to 10,000 rpm. The only failure experienced with this configuration happened when lubrication to the cone-rib interface was interrupted. The operation temperature was at all times very acceptable with very low oil flow rates. The power losses were lower than those for Configuration I.

Oil Supply to the Bearings

As mentioned before, the configurations are equipped for circulating oil lubrication. Figures 3.2 and 3.3 show the oil inlet and outlet points for each configuration. Configuration I, is provided with two oil inlets per bearing, one at each side of the spindle housing. Once inside the housing the oil is forced around the bearings through a groove in the outer surface of the outer race the bearing. The oil enters the bearing through three holes in the outer ring 120 degrees apart, provided for that purpose. Through these holes the oil is forced into the bearing cavity between the two rows of elements as shown in Figure

3.6. The oil is then forced out of the bearings, by the rolling element motion and centrifugal forces. The oil is then sucked out of the bearings through the exit ports.

Configuration II is provided with three oil inlet points: two for the high speed bearing, and one for the HYDRA-RIB_{TM}. Once the oil enters the housing it is directed to the front of the bearings by the distribution ring. Both bearings are provided with rings at the front (small end of the rollers). The high speed bearing is provided with a second distribution ring which feeds three oil jets. These jets direct the flow to the back of the cone, which is provided with a special ring. This ring entraps the oil from the jets, which is then fed centrifugally into the cone-rib interface through holes drilled for that purpose in the cone.

Configuration II was designed for horizontal use. When mounted in the vertical position, the upper bearing does not receive the required lubrication due to gravitational forces. Since the oil is sprayed up from the distribution ring, in vertical applications, it does not have the necessary pressure to force the oil through the bearing. To correct this problem, a screw type pump was provided above the HYDRA-RIB_{TM}. This pump supplied the necessary pressure drop to overcome gravity and provide an efficient flow of oil as long as a supply of 3.8 lpm is maintained to the top bearing.

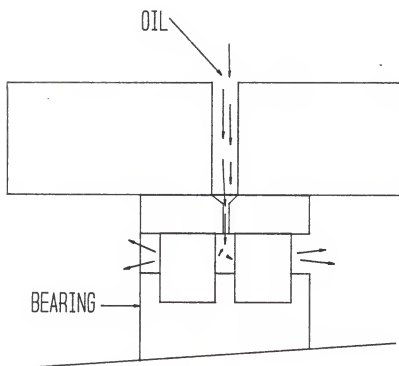


Figure 3.6 Oil Supply to Double Row Bearings

Instrumentation

During the tests of configurations I and II, the temperatures were monitored using type K thermocouples placed at strategic positions in the test rig. The thermocouples were connected to a digital display thermometer. The thermocouples were located at the following positions:

1. In the oil supply line.
2. In the oil return line.
3. At the oil exit point of each bearing.
4. At the outer race of the bearings.
5. On the surface of the housing.

The thermocouples at 1. and 2. measured the bulk oil temperatures before and after passing through the housing. The thermocouples used at 3. were in the suction line removing oil from each bearing. These thermocouples measured the exit temperature of the oil from each bearing, while 2. measures the temperature of the mixture of the oil from all bearings. Position 4. was measured for each bearing through a hole in the housing. Position 5. was measured at surface points above position 4. The thermocouples used in 1., 2., and 3. were in direct contact with the oil. The thermocouples used in 4. were encapsulated in a bayonet type assembly. The thermocouples used in 5. were in direct contact with the housing.

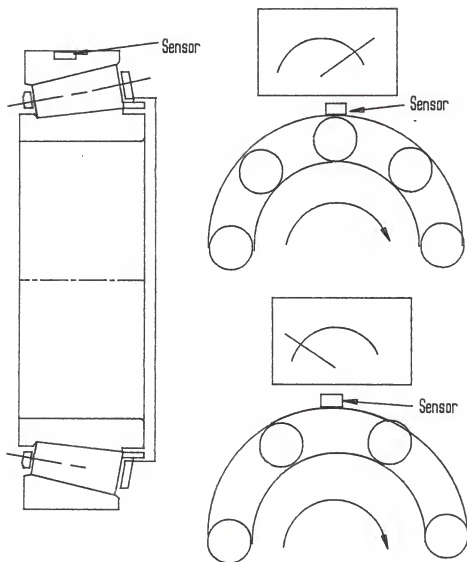


Figure 3.7 PROMESS Sensor

The load on the bearings was monitored using the PROMESS sensor. As shown in Figure 3.7, strain gages are located on the outer surface of the outer ring. As the loaded elements pass over the strain gages, these will provide an electrical signal proportional to the rolling element load. The PROMESS sensor is especially useful when monitoring the transient loads on the bearings.

The spindle speed was measured by using a magnetic pickup and gear installed at the top of the spindle. The speed was displayed on a electronic counter at all times. This speed was compared against the speed measured using a handheld tachometer. The speed was monitored throughout the test.

The input power to the motor was monitored using a set of current and voltage meters in the motor controller box. These meters measured the current and voltage supplied to the D.C. motor. The input power was computed from these measurements.

Oil Circulating System

There are several lubrication methods used in machine tools among them, grease lubrication, oil mist lubrication, air-oil lubrication ("O-L"), and circulating oil lubrication. Although the amount of oil required for lubrication is small, for high speed applications large amounts of oil must be used to provide the bearings with the

necessary cooling. The oil circulating through the bearing cavity removes a large part of the heat generated. So far, circulating oil is the only lubrication method which provides the necessary cooling for high speed bearing applications.

The oil used throughout the investigation was a SAE 10 equivalent oil, common in machine tools. A single type of oil was used. The use of a heavier oil will increase the hydraulic power losses and consequently, the operating temperature of the bearings. The power available for useful work (milling) will also be reduced due to an increase in hydraulic power losses. If on the other hand a lighter oil is used, the oil may exceed its operating range at high speeds and degrade. The friction between the rolling elements and the raceways would then increase, inducing an even larger operating temperature.

Figure 3.8 shows the circulating loop for the cooling and lubrication of the housing. Since circulating oil lubrication is going to be used to cool the bearings, large quantities of oil are necessary. The oil must be kept at constant temperature, since the experimental investigation would be affected by a variable supply oil temperature. The oil is pumped from a 280 l storage tank to the spindle by the supply pump. The supply pump is a variable vane pump with an operating range from 4 lpm to 53 lpm. Just before reaching the spindle, the oil flow is distributed into three streams. Each stream is controlled by a combination of

needle valve and a flow meter. Here, the amount of oil going into each bearing is measured and controlled. If configuration I, is being tested, each of the three streams is then split in two, to supply the oil to the bearing from both sides of the housing.

Once the oil has circulated through the bearings, removing heat from the bearing cavity, it is sucked out of the housing and returned into the storage tank by the suction pump. Due to the amount of churning within the bearings, the oil exiting the spindle is sucked out as foam. In the storage tank the oil is defoamed and cooled. To defoam the oil, it is passed through the screens, which removes the entrapped air. The oil is then pumped from the tank through the heat exchanger by the circulating pump. The cooling fluid in the heat exchanger is chilled water, from the laboratory's air conditioning system. After passing through the heat exchanger, the cold oil is returned back to the storage tank, near the warm oil return point, refer to Figure 3.8.

It is a known fact that the larger the difference in temperature between the two fluids in the heat exchanger, the more efficient it works. The need to remove the foam from the oil before it passes through the heat exchanger limits the alternatives as where to locate the inlet to the heat exchanger. If the suction point of the cooling circuit is placed next to the warm oil return, all the foam coming into the tank will be pumped into the heat exchanger,

reducing its efficiency. Therefore, the suction of the cooling circuit must be placed on the proper side of the screens, the closest possible to the warm oil return.

Evaluation of Cooling Capacity

In the initial stages of the investigation, it was observed that the temperature of the supply oil increased during the test. This increase in temperature significantly affected the investigation since the bearing temperature could not be related to a constant oil supply temperature. Therefore, an evaluation of the cooling system was performed. The question to be answered was if the circulating system was capable of providing the necessary cooling effect, removing from the warm oil all the heat it acquired from the bearings.

The amount of heat removed by the oil, from the spindle is given by

$$Q_{oilB} = m \rho c (T_{out} - T_{in}) \quad (kW) \quad (3.1)$$

where

Q_{oilB} : Heat removed by the oil from the bearings
(kW).

m : Oil flow rate (lpm).

ρ : Oil density (g/ml).

c : Heat capacity of the oil (kJ/(kg °C)).

T_{in} : Supply oil temperature (°C).

T_{out} : Return oil temperature (°C).

The heat removed from the oil in the heat exchanger is given by

$$Q_{oilHE} = m \rho c (T_{in} - T_{out}) \quad (kW) \quad (3.2)$$

where

Q_{oilHE} : Heat removed from the oil (kW).

m : Oil flow rate (lpm).

ρ : Oil density (g/ml).

c : Heat capacity of the oil (kJ/(kg °C)).

T_{in} : Oil temperature entering the heat exchanger (°C).

T_{out} : Oil temperature exiting the heat exchanger (°C).

The heat removed by the water in the heat exchanger is given by

$$Q_{waterHE} = m \rho c (T_{out} - T_{in}) \quad (kW) \quad (3.3)$$

where

$Q_{waterHE}$: Heat removed by the water from the oil (kW).

m : Cooling water flow rate (lpm).

ρ : Density of the water (g/ml).

c : Heat capacity of the water (kJ/(kg °C)).

T_{in} : Water temperature entering the heat exchanger (°C).

T_{out} : Water temperature exiting the heat exchanger (°C).

Experimental data was collected at steady state, it is listed in Table 3.1. With this data, the amount of heat

removed from the oil in the heat exchanger, the amount of heat acquired by the oil from the bearings and the amount of heat acquired from the oil were computed. It was found that the heat exchanger did have the necessary capacity to cool the oil to the desired supply temperature. As it can be observed from Table 3.1, the temperature of the oil entering the heat exchanger is much lower than the temperature of the returning oil. Therefore, the problem was not that the heat exchanger could not supply the necessary cooling, but that the warm oil was not getting to the heat exchanger until it is too late. Upon inspection of the tank, it was found that the oil inlet to the heat exchanger was too far from the warm oil return point. This caused the warm oil to concentrate on one side of the tank, heating that side of the tank. This accumulation of warm oil increased until it reached the heat exchanger oil inlet. By that time, the amount of oil which needed to be cooled was above the cooling capacity of the heat exchanger, which in the mean time was circulating cool oil.

To solve the problem the oil inlet point into the heat exchanger was moved closer to the oil return point. It could not be moved close enough since it must be placed after the screens, otherwise, the foam would make its way into the heat exchanger, reducing its cooling capacity. The final solution was to return the cold oil beside the warm oil return. This kept the return side cold and there was no chance for the warm oil to accumulate in that side.

Table 3.1 Heat Exchanger Temperatures

Speed	Oil Flow	Water		Oil		Ret.Oil
		In	Out	In	Out	
3000	1.9 lpm	10°C	16°C	31°C	13°C	44°C
3000	3.0 lpm	11°C	17°C	33°C	14°C	41°C
3000	3.8 lpm	10°C	18°C	33°C	15°C	38°C
4000	0.8 lpm	8°C	12°C	14°C	12°C	66°C
4000	1.9 lpm	11°C	18°C	36°C	21°C	53°C
4000	3.0 lpm	11°C	20°C	37°C	23°C	49°C
4000	3.8 lpm	11°C	21°C	37°C	22°C	49°C
5000	1.9 lpm	10°C	22°C	26°C	22°C	66°C
5000	3.0 lpm	10°C	25°C	31°C	25°C	62°C
5000	3.8 lpm	9°C	27°C	33°C	27°C	61°C
6000	0.8 lpm	7°C	11°C	12°C	10°C	64°C
6000	1.5 lpm	6°C	12°C	16°C	12°C	67°C

Seals

As both configurations are lubricated using circulating oil and mounted in the vertical position, proper sealing is imperative. Any oil that leaks out of the housing, through the bottom, will fall on the workpiece. This oil may affect the life of the tool by exaggerating the thermal cycling of the tool, causing the failure of the tool. Also, it represents a hazard to the operator, since at high speeds, the oil is sprinkled onto the surroundings, making the area quite slippery.

Due to the high rotational speeds, non-contact seals must be used. Non-contact seals have the extra advantage that they do not contribute to the friction torque, thus reducing the amount of heat generated. A similar arrangement of labyrinth seal was used for both configurations. A section view of the seal, for Configuration I and Configuration II, is shown in Figure 3.9 and in Figure 3.10, respectively. Both configurations were effectively sealed for most of our operating conditions.

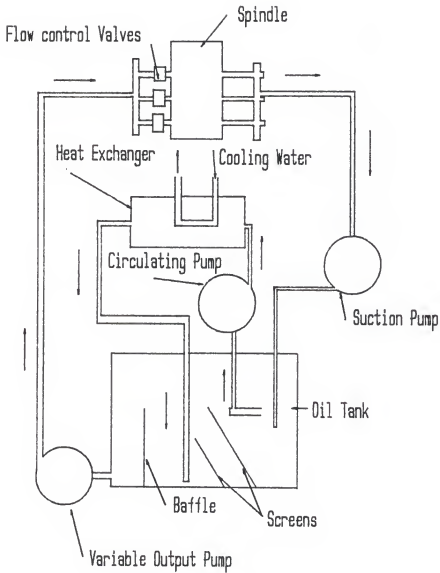


Figure 3.8 Circulating Oil System

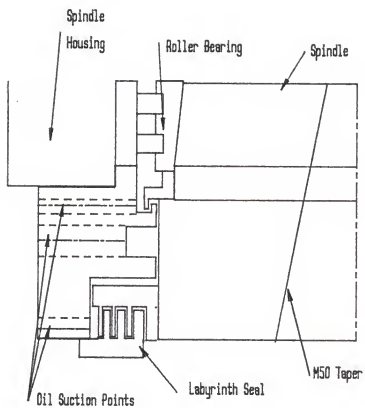


Figure 3.9 Seal for Configuration I

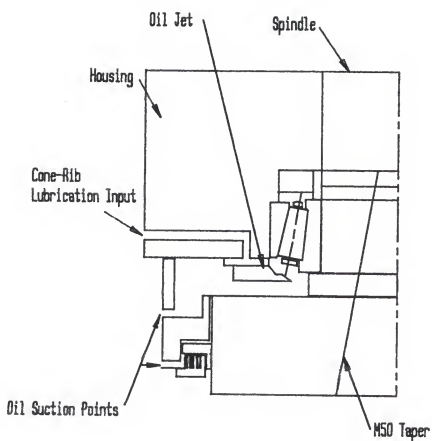


Figure 3.10 Seal for Configuration II

CHAPTER IV

THERMAL ANALYSIS

Thermal Analysis of the Spindle Housing

To estimate the heat generation rates of bearings, researchers and bearing manufacturers have developed several empirical and theoretical equations. These equations relate heat generation to bearing geometry, operating conditions and lubrication parameters. In this chapter these relationships will be presented and compared among themselves and to experimental results.

Also in this chapter, thermal profiles are presented, showing temperature distribution along the housing. The presence of thermal gradients between the bearings and the spindle housing may induce an increase of the original preload, which in some instances may cause bearing seizure. The thermal gradient is induced by the faster increase in rolling element temperature compared to the housing during the acceleration of the spindle to the operating speed.

Friction in Rolling Bearings

The heat generated in the bearings is the product of frictional power losses. The sources of these frictional losses as identified in [16,29,30] are:

1. Elastic hysteresis in rolling. As the bearing rolls there are deformations in the raceways and in the rolling elements. The energy consumed in producing this deformation is partly recovered when the element rolls to the next position.
2. Sliding in rolling-element/raceway contacts due to the geometry of the contacting surfaces.
3. Sliding due to deformation of contacting elements.
4. Sliding between the cage and the rolling elements, and between the cage and the guiding surfaces.
5. Sliding between roller ends and inner and/or outer ring flanges.
6. Viscous drag of the lubricant on the rolling elements and cage. The viscous friction is produced by the internal friction of the lubricant between the working surfaces. Also the churning of the oil between the cage and the rolling elements, between the raceways and the rolling elements and flanges. These losses increase with speed and amount of lubricant in the bearing cavity.

In the experimental investigation, the effect of the above power losses were grouped into two measurable amounts, Mechanical Power losses and Hydraulic Power losses. The

mechanical power losses are the consequence of mechanical friction in the bearing cavity, without oil being circulated through the bearing. The hydraulic power losses are the results of viscous friction between the oil in the bearing cavity and the rolling elements. These two main sources of heat are discussed in Chapter V.

Heat Generation

The increase in temperature during the operation of the bearings is the result of friction losses, which are manifested as heat. The sources of friction in a bearing, as mentioned above, include the friction at the contact between rolling element and each raceway, friction between the cage and the rolling elements and viscous drag between the circulating oil and the rolling elements. Several empirical relations have been developed to estimate the amount of heat generated in a bearing.

The frictional power consumed by a bearing is given by [9,11] as

$$H_f = 1.05 \times 10^{-4} n M \quad (W) \quad (4.1)$$

where

H_f : heat generated (W)

n : spindle speed (rpm)

M : friction torque (N-mm)

Also from [9,11], the bearing manufacturer estimates the friction torque as

$$M = 0.5 \mu_f F d \quad (\text{N-mm}) \quad (4.2)$$

where

μ_f : friction coefficient for the bearing

F: bearing load (N)

d: bore diameter of the bearing (mm)

The friction coefficient μ_f is given in [9,11] for several types of bearings

for cylindrical roller bearings $\mu_f = 0.0011$

for thrust ball bearings $\mu_f = 0.0013$

for tapered roller bearings $\mu_f = 0.0018$

These friction coefficients are for single row bearings operating at average speed and at a load for a life of 1000 million revolutions.

The loads acting on the bearings are reactions to the belt tension. The magnitude of this tension is computed following the procedure suggested by the manufacturer in [31] for the type of belt used. For Configuration I, the tension load is 8600 N, while for Configuration II, the tension load is 3600 N. With the tension load known and using load equilibrium, the bearing reactions for configuration were determined. For Configuration I, the load on the lower bearing (NN3022 K) was estimated at 3000 N, for the top bearing (NN3019 K) it was 12000 N and the center bearing, the Radiax, 400 N which is the weight of the spindle. For Configuration II, the load acting on the lower

bearing, the High Speed Bearing, was estimated as 1600 N, while at the Hydra-RibTM the belt tension component was 5200 N and an axial component of 400 N due to the weight of the spindle.

Using equation (4.2) to compute the friction torque for both configurations, using double the friction coefficient for the double row bearings, the following estimates were obtained:

for Configuration I

$$M_{NN3022K} = 360 \text{ N-mm}$$

$$M_{NN3019K} = 1232 \text{ N-mm}$$

$$M_{234420BFI} = 46 \text{ N-mm}$$

for Configuration II

$$M_{HYDRA-RIB} = 489 \text{ N-mm}$$

$$M_{HS} = 146 \text{ N-mm}$$

The heat generated, computed using equation (4.1), at the different test speeds, for each configuration are listed next.

Configuration I

$$@ 3,000 \text{ rpm} \quad 516 \text{ Watts}$$

$$@ 5,000 \text{ rpm} \quad 859 \text{ Watts}$$

$$@ 7,000 \text{ rpm} \quad 1204 \text{ Watts}$$

$$@ 8,000 \text{ rpm} \quad 1376 \text{ Watts}$$

Configuration II

$$@ 3,000 \text{ rpm} \quad 781 \text{ Watts}$$

$$@ 5,000 \text{ rpm} \quad 1312 \text{ Watts}$$

@ 7,000 rpm 1837 Watts

@ 9,000 rpm 2362 Watts

A more accurate way to compute the friction moment is by dividing it into two parts: an idling torque M_o and a load torque M_1 . The sum of the two is the friction torque. The idling torque represents the friction torque during idle operation of the bearing and is given by [9,11] as

$$M_o = f_o \times 10^{-8} (vn)^{2/3} d_m^3 \quad vn \geq 2000 \quad (4.3)$$

$$M_o = f_o \times 160 \times 10^{-8} d_m^3 \quad vn < 2000 \quad (4.4)$$

where

f_o : factor depending on bearing design and
lubrication method, for vertical spindles and
oil jet lubrication:

for double row ACBB..... 9

for CRB 4-6

for TRB 8-10

v : oil viscosity at working temperature (cS)

d_m : mean diameter of the bearing (mm)

The friction torque due to the applied load can be computed using an equation recommended by Palmgren, [11].

$$M_1 = f_1 F_B d_m \quad (N\text{-mm}) \quad (4.5)$$

where

M_1 : friction torque due to the load (N-mm)

f_1 : factor dependent on the geometry of the
bearing and relative load.

F_B : equivalent force, as described below (N)

d_m : mean bearing diameter (mm)

Recalling equation (2.2), for ball bearings, the factor f_1 is given by

$$f_1 = z \left(\frac{F_o}{C_o} \right)^y \quad (4.6)$$

for angular contact ball bearings, $z=0.0001$ and $y=0.33$ [11].

For roller bearings, f_1 will be

for cylindrical roller bearings:

$$f_1 = 0.0002 - 0.0003$$

for tapered roller bearings:

$$f_1 = 0.0003 - 0.0004$$

F_s for ball bearings is given by the following equations, also from [11].

$$F_s = 0.9F_a \tan \alpha - 0.1F_r \quad (4.7)$$

or

$$F_s = F_r \quad (4.8)$$

whichever is larger, (4.7) or (4.8).

For radial roller bearings, F_s is given below as

$$F_s = 0.8F_a \tan \alpha \quad (4.9)$$

or

$$F_s = F_r \quad (4.10)$$

whichever is larger, (4.9) or (4.10).

In Figures 4.1 and 4.2, the computed generated heat is plotted at different test speeds and oil flow rates for Configurations I and II, respectively. The generated heat was computed by adding the idle friction torque and the applied load friction torque and substituting the sum into

equation (4.1). In Figures 4.3 and 4.4, the power losses determined experimentally for Configurations I and II, respectively, are plotted. The experimental power losses shown in the figure represent the sum of the Mechanical Power Losses and the Hydraulic Power Losses, which are defined in Chapter V. As it can be observed by comparing Figures 4.1 and 4.2 against Figures 4.3 and 4.4, equations (4.2) to (4.3) predicted a heat generation much lower than the measured during the test. The supply oil temperature and the return oil temperature were used to compute an average oil temperature for the computation of the viscosity of the oil inside the bearing cavity.

For their tapered bearings, TIMKEN recommends in [10] the equations that follow to estimate the friction torque and the heat generation.

$$M = k_1 G (S\mu)^{-5} (F_{eq})^{1/3} \quad (N\cdot m) \quad (4.11)$$

where

M: bearing operating torque (N·m)

k_1 : conversion factor = 7.56×10^{-6} (metric units)

G: bearing geometry factor as given in the TIMKEN bearing catalog [32,33].

for HYDRA-RIB_{TM} = 152.7

for High Speed Bearing = 129.5

S: spindle speed (rpm)

μ : oil viscosity (Centipoise)

F_{eq} : equivalent axial load (N)

if the bearing is under combined loading, the equivalent

load F_{eq} is determined as

$$\text{if } \frac{K}{F_r} * F_a > 2.5 \text{ then } F_{eq} = F_a \quad (4.12)$$

otherwise

$$F_{eq} = f_c F_r \frac{1}{K} \quad (N) \quad (4.13)$$

where

F_a : axial load (N)

F_r : radial load (N)

K : bearing K factor, from the TIMKEN bearing catalog [32]:

for HYDRA-RIB_{TM} = 1.63

for High Speed Bearing = 1.23

f_c : axial load factor, function of $(KF_a)/F_r$ as given in the bearing catalog [33].

To compute the heat generation rate for tapered roller bearings, equation (4.14), which is recommended by TIMKEN for their bearings was used. The computed generated heat is plotted in Figure 4.5, versus spindle speed at constant oil flow rate.

$$Q = k_2 G S^{1.5} \mu^{.5} F_{eq}^{-1/3} \quad (4.14)$$

where

Q : heat generation (W)

k_2 : conversion factor (metric) = 7.9×10^{-7}

As it can be observed by comparing Figure 4.5 against the experimental measurements in Figure 4.4, equation (4.14) predicted quite well the generated heat for Configuration II.

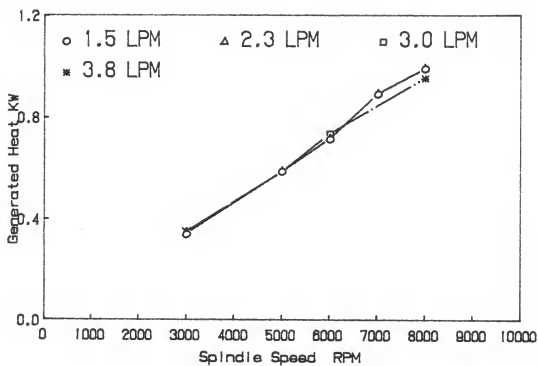


Figure 4.1 Computed Generated Heat vs. Spindle Speed
Configuration I

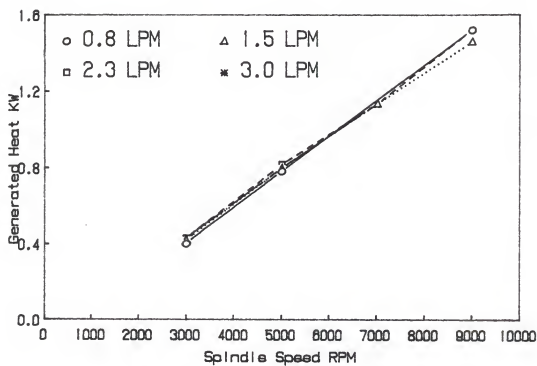


Figure 4.2 Computed Generated Heat vs. Spindle Speed
Configuration II

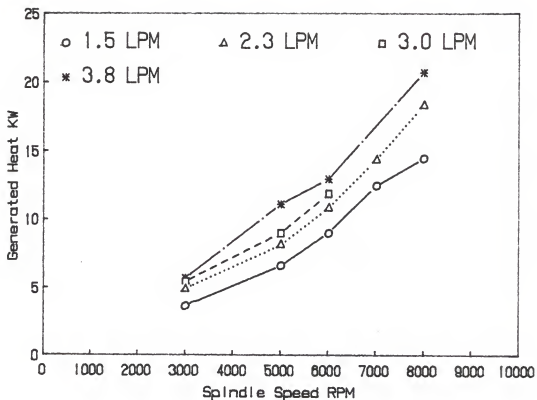


Figure 4.3 Experimentally determined Heat Generation
Configuration I

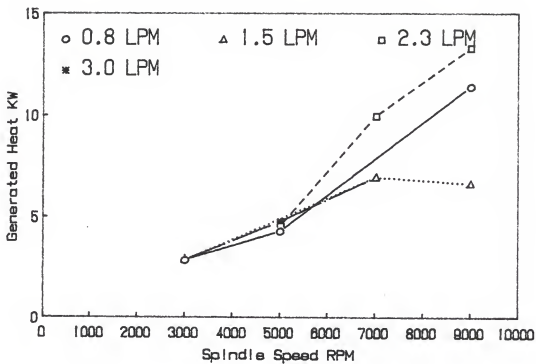


Figure 4.4 Experimentally Determined Heat Generation
Configuration II

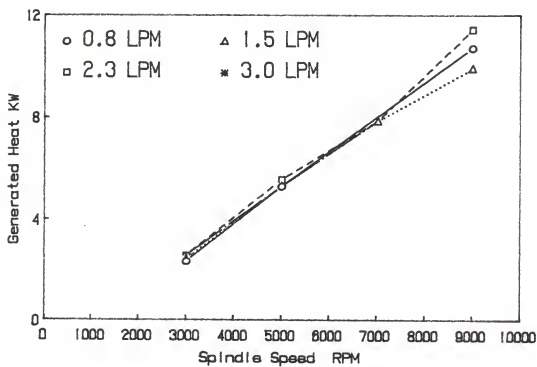


Figure 4.5 Computed Heat Generation
Configuration II

Heat Removal

During high speed operation of rolling elements, the heat generated within the bearings is considerable, as it will be shown in Chapter V. This heat must be removed to avoid excessive thermal loads on the elements. If the temperature rises too much, the lubricant may exceed its operating range and the oil film between raceways and elements could be eliminated.

Circulating oil lubrication has the largest heat removal capacity, due to the amount of oil which is forced through into the bearing cavity. The amount of heat removed by the oil can be computed by multiplying the mass flow of the circulating oil, by its heat capacity, times the change in temperature.

$$P_{oil} = (m\dot{q}c)_{oil}(T_{out}-T_{supply}) \quad (\text{kW}) \quad (4.15)$$

where

P_{oil} : power removed by the oil (kW)

m : oil flow rate (lpm)

$\dot{q}c$: specific heat times the density of the oil,
1566 (KJ/(m³ C°))

T_{out} : oil temperature at the exit of the housing
(°C)

T_{supply} : oil temperature at the inlet of the
housing (°C)

Table 4.1 Removed Heat/Generated Heat
Configuration I

Speed RPM	Oil Flow Rate (LPM)	Generated Heat (kW)	Removed Heat (kW)	Percentage Removed
3000	1.5	3.6	2.7	76
3000	2.3	4.9	4.0	81
3000	3.0	5.5	4.5	82
3000	3.8	5.6	5.0	90
5000	1.5	6.6	4.6	70
5000	2.3	8.1	6.3	78
5000	3.0	9.0	7.0	78
5000	3.8	11.1	9.2	83
6000	1.5	9.0	5.4	61
6000	2.3	10.8	7.4	68
6000	3.0	11.8	9.2	77
6000	3.8	12.9	10.7	83
7000	1.5	12.5	5.26	42
7000	2.3	14.3	8.3	58
8000	1.5	14.4	5.8	40
8000	2.3	18.3	9.4	51
8000	3.8	20.7	17.3	83

Table 4.2 Removed Heat/Generated Heat
Configuration II

Speed RPM	Oil Flow Rate (LPM)	Generated Heat (kW)	Removed Heat (kW)	Percentage Removed
3000	0.8	2.9	2.0	70
3000	1.5	2.9	2.4	82
3000	3.0	2.9	2.7	93
3000	3.8	2.9	2.6	89
5000	0.8	4.3	3.4	80
5000	2.3	4.6	3.6	79
5000	3.0	4.8	3.9	82
5000	3.8	4.8	4.0	82
7000	1.5	7.0	4.8	64
7000	2.3	7.0	4.9	70
7000	3.0	7.0	5.5	79
9000	0.4	10.6	5.5	52
9000	0.8	11.4	6.0	53
9000	1.5	12.4	6.3	54
9000	2.3	13.3	7.7	58

As the amount of oil increases, so does the cooling capacity, removing more heat from the bearings. The ratio of the heat removed to the heat generated increases with increased oil flow rate. In Tables 4.1 and 4.2, the percentages of generated heat removed by the oil are listed, for each configuration, at each speed and oil flow rate. As it can be seen from the table, as the oil flow rate increases, the percentage of the generated heat which is removed increases. Also from the table, as speed increases, for the same flow rate, the percentage of the generated heat removed by the oil decreases. This can be explained by the fact that as the speed increases, so does the temperature of the bearing, as it will be shown later. A higher bearing temperature will produce a larger heat conduction rate through the housing due to a larger temperature gradient between the housing and the environment. Thus, less heat is convected away by the oil. Also with an increase spindle speed, the oil in the bearing cavity traps a larger amount of air, changing itself into foam and hence reducing its convection capacity.

An increase in oil flow rate will also produce an increase in power losses, as it will be shown in Chapter V. The increase in power losses is, in some cases, large enough to nullify the increased cooling capacity that a larger oil flow rate produces. Therefore, the net effect may be an insignificant decrease in temperature and a significant increase in power losses. From the experimental results,

such as power losses, operating temperature and oil flow rate, design recommendations will be made for each configuration.

Steady State Temperature Fields

The steady state thermal fields were computed for both configurations. The analysis was performed using finite difference methods. The housing was divided into ring elements as shown in Figure 4.6 for Configuration II. The initial temperature was taken as room temperature except for those elements covered by the boundary conditions. The program was stopped when the surface temperature of the model approximated the experimentally measured surface temperature. Forced convection at the housing surfaces was assumed since the spindle rotation produces a considerable flow of air around the spindle. The equations used to estimate the amount of heat conducted radially from one element to the next are given by [34] as

$$Q_{\text{radial}} = \frac{2\pi k l}{\ln \frac{R_o}{R_i}} * (T_3 - T_1) \quad (\text{Watts}) \quad (4.16)$$

where

Q_{radial} : Heat transmitted in the radial direction
(Watts)

k: thermal conductivity of the housing material

(cast iron= 52 W/(m²°C) [34]

l: axial length of the element (m)

T_j: temperature of the jth element (°C)

T_i: temperature of the ith element (°C)

R_o: outer radius of the housing (m)

R_i: inner radius of the housing (m)

The equation used to compute the heat conducted from one element to the next in the axial direction is

$$Q_{\text{axial}} = \frac{2\pi k r dr}{x} (T_j - T_i) \quad (\text{Watts}) \quad (4.17)$$

where

Q_{axial}: Heat transmitted in the axial direction
(Watts)

r: radius of the ith element (m)

dr: radial width of the element (m)

x: axial distance between nodes (m)

The equation used to estimate the heat conducted away by the air surrounding the spindle housing is given by [34] as

$$Q_{\text{convection}} = h A (T_h - T_a) \quad (\text{Watts}) \quad (4.18)$$

where

Q_{convection}: Heat removed away by convection
(Watts)

h: convection coefficient = 9 W/m² [34]

A: heat transfer area (m²)

T_h: housing surface temperature (°C)

T_a : temperature of the surroundings ($^{\circ}\text{C}$)

Equation (4.19) was used to estimating the radiation heat transfer.

$$Q_{\text{radiation}} = \sigma \epsilon F A (T_h^4 - T_a^4) \text{ (Watts)} \quad (4.19)$$

where

$Q_{\text{radiation}}$: Heat removed away by radiation (Watts)

σ : Stefan-Boltsman constant = 5.66961×10^{-8}

($\text{W}/(\text{m}^2\text{K}^4)$) [34]

ϵ : emissivity (.8) [34]

F : shape factor = 1.0 [34]

The boundary conditions used for the analysis of each housing were:

1. The bearings are represented as elements with constant temperature. The temperature assigned is the temperature of the bearing at steady state measured in the test.
2. The temperature at the inside surface of the housing is assumed to be equal to the average between the surface temperature of the center and the average bearing temperature, for the given speed and oil flow rate.
3. At the outer surface the housing loses heat to the environment through convection and radiation.
4. There is forced convection and radiation at the top surface.

5. The temperature of the environment was assumed constant at 22°C.

The computed thermal profiles for Configuration II at 5,000 rpm, 7,000 rpm and 9,000 rpm, and an oil flow rate of 2.3 lpm are shown in Figures 4.7, 4.8 and 4.9. The computed thermal profiles will be used to compute thermally induced loads on the rolling elements.

Thermally Induced Loads

As heat is generated in the bearings, a temperature gradient is developed between the bearings and the outer surface of the housing. Since the bearings and the housing are heating at different rates, their expansions occur at different rates. These differential expansions induce loads on the bearings. These loads will be proportional to the difference in thermal expansions between the bearings and the housings.

Let's assume that the inner race, the rolling elements and the outer race are all at the same temperature. The thermal expansions of the inner ring, the outer race and the i th ring of the housing model are given respectively by

$$\delta_{Ti} = \Gamma d_i \pi (T_i - T_m) \quad (m) \quad (4.20)$$

$$\delta_{To} = \Gamma d_o \pi (T_o - T_m) \quad (m) \quad (4.21)$$

$$\delta_{Thi} = \Gamma d_{hi} \pi (T_{hi} - T_m) \quad (m) \quad (4.22)$$

where

δ_{Ti} : thermal expansion of the inner ring (m)

δ_{T_o} : thermal expansion of the outer ring (m)

$\delta_{T_{h1}}$: thermal expansion of the i^{th} housing element
(m)

Γ : thermal expansion coefficient $10.6 \times 10^{-6} \text{ } ^\circ\text{C}^{-1}$
[30]

d_i : inner ring diameter (m)

d_o : outer ring diameter (m)

d_{h1} : diameter of the i^{th} housing element (m)

T_i : temperature of the inner ring ($^\circ\text{C}$)

T_o : temperature of the outer ring ($^\circ\text{C}$)

T_{h1} : temperature of the i^{th} housing element ($^\circ\text{C}$)

T_s : starting temperature ($^\circ\text{C}$)

The thermal expansion of the outer race is prevented by the much slower expansion of the housing. It is at this joint that the thermally induced interference happens, increasing the bearing preload. To determine the induced load, the expansion of the housing must first be computed. Using the thermal fields computed above, the expansion of each ring element in the housing can be computed. The expansion of the element in contact with the bearing can therefore be computed, and after computing the expansion of the outer ring of the bearing, the increase in interference could be determined. By using the load-deflection relationships developed in Chapter VI, the thermally induced load could be computed.

Computation of Thermal Loads

Following the procedure described above for computing the thermal loads, a sample calculation will now be provided for the 7,000 rpm test of Configuration II. The thermal deflection at each concentric ring surrounding the lower bearing is first computed using the temperature distribution as shown in Figure 4.8. The outer diameter of the bearing element is 0.158 m. The next element is .012 m larger and the rest are divided using 0.026 m increments. Using equations (4.20) to (4.22), the thermal deflections are computed next using T_m as 295°K.

For the outer race element, the thermally induced deflection is

$$\delta_{To} = 10.6 \times 10^{-6} * (0.158) * \pi * (340 - 295) \quad (\text{m})$$

$$\delta_{To} = 2.37 \times 10^{-4} \quad (\text{m})$$

This δ_{To} would be the deflection of the outer ring if it was not constrained by the other ring elements. To determine then the actual deflection, the deflections of all the rings must be computed. Once the thermal deflection of the outermost ring is estimated the deflection of the outer race of the bearing is determined. The minimum deflection computed for any of the rings surrounding the outer race was of 2.26×10^{-4} m. Thus, the maximum deflection of the outer race of the bearing is that of the ring which deflected the less or 2.26×10^{-4} m.

To determine the increase in preload, the thermal deflection of a roller must first be computed. It is given

by equation (4.22) using the diameter of the roller instead of the element diameter.

$$\delta_{\tau o} = 10.6 \times 10^{-6} * (0.013) * \pi * (340 - 295) \quad (\text{m})$$

$$\delta_{\tau o} = 2.37 \times 10^{-4} \quad (\text{m})$$

The increase in preload can now be estimated by subtracting the roller thermal deflection from the outer ring thermal deflection. This difference is multiplied by the stiffness of the bearing to obtain the increase in load. Thus, the difference in thermal deflections is 1.1×10^{-5} m. Using equation (6.12) and a roller stiffness value of 1.00×10^8 N/m [30] the load was computed as 3.1×10^2 N. This load is negligible for the type of bearing used. This coincides with the PROMESS sensor measurements.

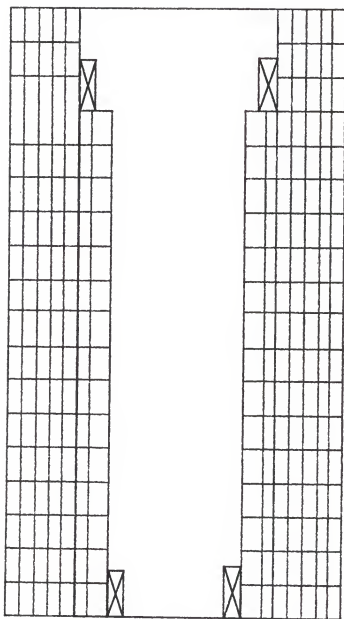


Figure 4.6 Thermal Model for Configuration II

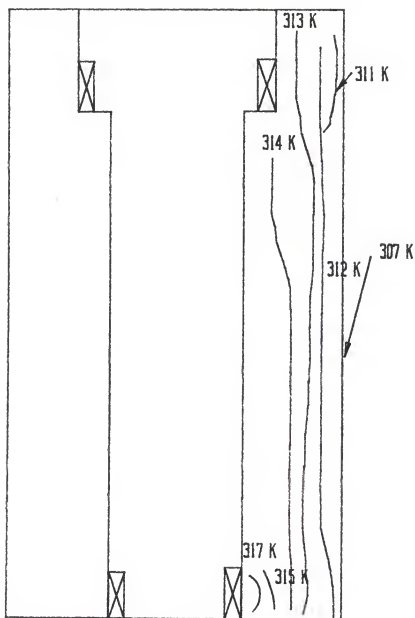


Figure 4.7 Thermal Profile (5,000 RPM)
Configuration II

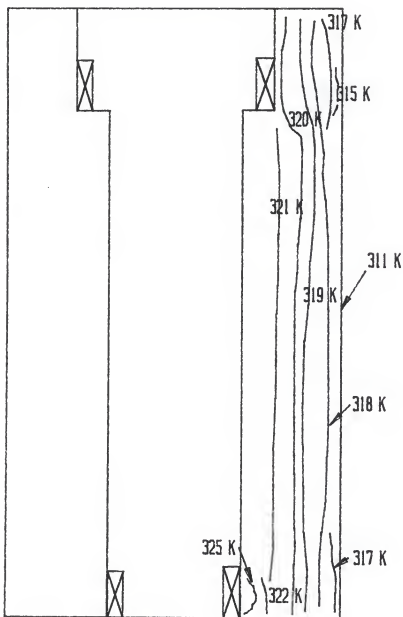


Figure 4.8 Thermal Profile (7,000 RPM)
Configuration II

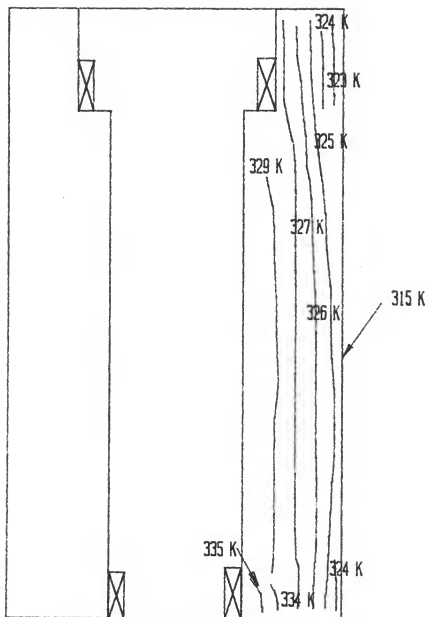


Figure 4.9 Thermal Profile (9,000 RPM)
Configuration II

CHAPTER V

EXPERIMENTAL RESULTS AND DISCUSSION

Test Procedure

The spindles were tested for performance, power losses, operating temperatures and maximum operation speed. The tests were run at idle, no load from a cutting process, on the spindle. The spindle was started at low speeds (300 rpm), the desired oil flow rate was pumped into the spindle and the initial bearing temperatures and loads were recorded. The spindle was then accelerated to the predetermined test speed. The test speed and oil flow rates were selected based on the performance of the spindle during previous tests.

During the transient stages, measurements were taken every minute. As the system stabilized, the measurements were taken at longer intervals, according to the changes in the test parameters. The parameters recorded during the test were:

1. Inlet Oil Bulk Temperature.
2. Outlet Oil Bulk Temperature.
3. Oil Temperature at the exit from each Bearing.
4. Outer Race Bearing Temperature.
5. Oil Flow Rate per Bearing.
6. Power imputed to the Spindle Drive System.

7. Spindle Speed.

8. Bearing Loads.

Curve Fitting of Experimental Data

The experimental data was used to develop empirical equations which describe the relationship between the independent variables spindle speed and oil flow rate, and to the dependent variables operating temperatures and power losses. It was found through the experimental investigation that for a given configuration, any combination of spindle speed and oil flow rate would determine a set of operating temperatures and corresponding power requirements. Using a public domain computer program developed by the IBM PC User Group of Greater Kansas City, the data was analyzed. The program uses the least square method to fit either a straight line, or a geometric curve or an exponential curve to a set of data inputted as x,y coordinates.

The results from the experimental investigation can be represented as tridimensional matrices. Two axis representing the independent variables spindle speed and oil flow rate, the third representing either operating temperatures or power losses. For each point where temperature was measured, for each configuration, such a matrix could be developed. As it will be demonstrated below, each configuration has its own behavior which characterizes its performance.

The computer program was used to develop relationships between one of the independent variables, the other was kept constant, and one of the dependent variables. The geometric relationship was selected since it was consistently the best fitting equation to the experimental data. The best fit was determined by observing the coefficient of determination (R^2) and the correlation coefficient (R) which are part of the program's output. The determination coefficient and the correlation coefficient are included with the curve fit constants in the discussion below.

Temperatures

The temperature of the bearings was recorded at different oil flow rates and speeds during both steady states and transient stages. A comparison can be made between the two configurations based on their operating temperatures given a spindle speed and oil flow rate. The temperatures of the supply oil, return oil and the lower bearing are plotted for several test speeds versus time, in Figures 5.1 to 5.9. These are the plots of the transient behavior of the configurations under investigation. It can be seen from these graphs the time it takes each configuration to achieve steady state. The effect of a different oil flow rate can also be observed from these graphs.

At steady state, the temperatures are plotted versus spindle speed and versus oil flow rate. An approximation function was derived from the experimental data using least squares regression analysis. The temperature of the bearings was the dependent variable, while spindle speed and oil flow rate were plotted as the independent variables.

Steady State Temperatures Versus Spindle Speed

Bearing temperatures for Configuration I and II are plotted vs. spindle speed at constant oil flow rate in Figure 5.10 and 5.11, respectively. Figure 5.10 shows the change in temperature of the NN 3022K/SP bearing, Configuration I, as the spindle speed is varied. The lower bearing was selected since it is the largest of the three bearings and the one with the highest operating temperatures. In Figure 5.11, the temperature of the High Speed Tapered Bearing, Configuration II, is shown. The oil flow rate to the Hydra-Rib_{TM} was kept constant at 3.8 lpm at all times. In the figures and throughout this chapter the experimental data is represented by symbols while the lines represent approximation functions derived from the experimental data. From Figures 5.10 and 5.11, a set of approximation functions were derived using a least square fit technique to estimate the temperature of the bearing for a given spindle speed at a constant oil flow rate. The functions are in the form

$$T_B = a_1 n^{a_2} \quad (^\circ\text{C}) \quad (5.1)$$

where

T_B : Bearing temperature ($^\circ\text{C}$)

n : Spindle speed (rpm)

a_1 : Curve fit constant one ($^\circ\text{C}/\text{rpm}^{a_2}$)

a_2 : Curve fit constant two

The value of a_1 and a_2 for Configurations I and II are given in Table 5.1 and Table 5.2. As it can be observed from Figure 5.10 and Figure 5.11, the temperature of the bearing increases with spindle speed. As the spindle speed increases, the heat generation also increases and consequently the higher operating temperatures. The steady state operating temperature of Configuration I reached 85°C at 7,000 rpm and 1.5 lpm. At 8,000 rpm and 1.5 lpm, the temperature reached 84°C without achieving steady state. To prevent bearing damage due to high temperature operation, the oil flow rate was increased.

The steady state operating temperature for Configuration II, reached 80°C at 9,000 rpm and 0.4 lpm. At all times the operating temperatures for Configuration II were much lower than for Configuration I, for the same speed and oil flow rate.

The approximation functions derived for each flow rate equation 5.1 with the constants in Table 5.1 and Table 5.2, match the experimental data very well, as it can be observed from the figures.

Steady State Temperatures Versus Oil Flow Rate

Figures 5.12 and 5.13 show the change in bearing temperature for constant speed and as the oil flow rate varies for Configurations I and II, respectively. Again, a set of approximation functions were derived to describe the effect of oil flow rate on bearing temperature. The equations are in similar form as in equation (5.1)

$$T_B = a_3 v^{a_4} \quad (^\circ\text{C}) \quad (5.2)$$

where

T_B : Bearing temperature ($^\circ\text{C}$)

v : Oil flow rate (lpm)

a_3 : Curve fit constant one ($^\circ\text{C}/\text{lpm}^{a_4}$)

a_4 : Curve fit constant two

The value of the constants a_3 and a_4 are given for each configuration at each test speed in Table 5.3 and Table 5.4. As it can be observed from Figure 5.12 and Figure 5.13, the temperature of the bearing decreases with oil flow rate. The decrease in bearing temperature with flow rate is to be expected, since by increasing the oil flow rate, the cooling capacity increases. Note that as the oil flow rate is increased the drop in temperature is smaller than with the previous increase in oil flow rate. This is due to the increase in power losses that an increase oil flow rate produces. As it was mentioned before, there is a compromise oil flow rate at which the bearing temperature is reasonable while the power losses are not that large.

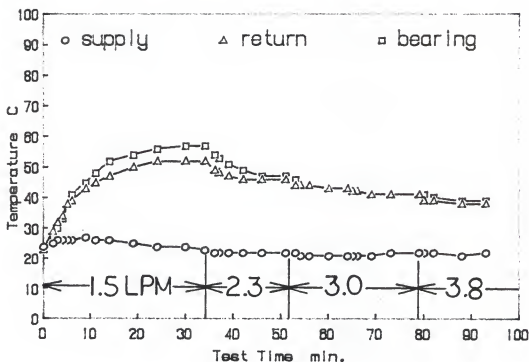


Figure 5.1 Transient Temperatures
Configuration I: 3,000 RPM

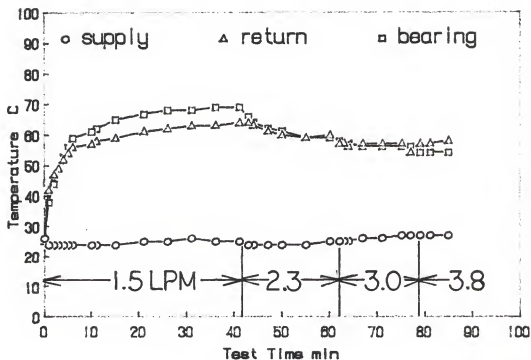


Figure 5.2 Transient Temperatures
Configuration I: 5,000 RPM

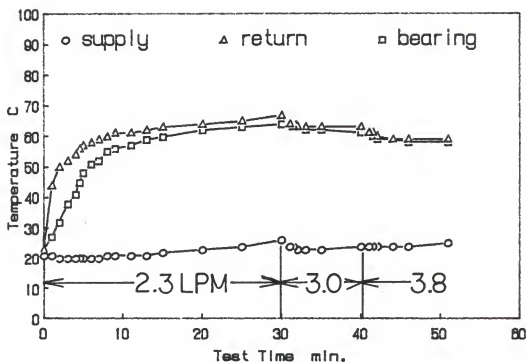


Figure 5.3 Transient Temperatures
Configuration I: 6,000 RPM

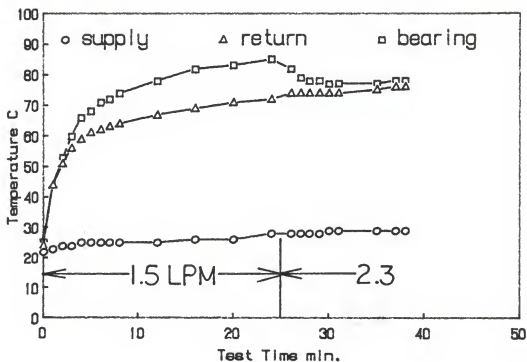


Figure 5.4 Transient Temperatures
Configuration I: 7,000 RPM

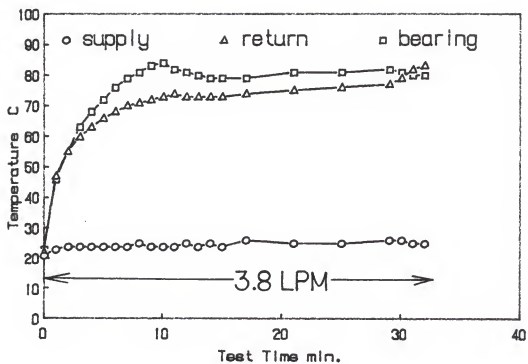


Figure 5.5 Transient Temperatures
Configuration I: 8,000 RPM

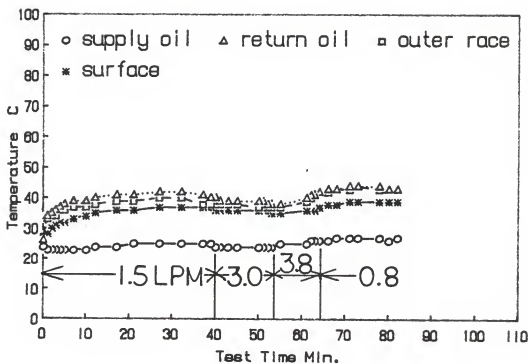


Figure 5.6 Transient Temperatures
Configuration II: 3,000 RPM

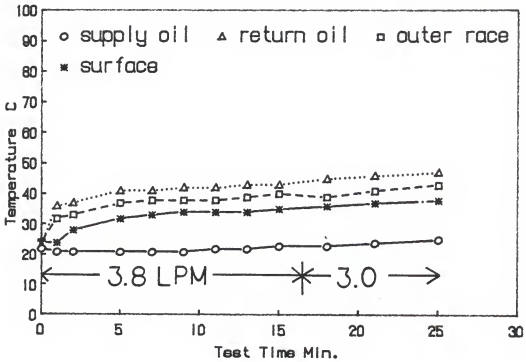


Figure 5.7 Transient Temperatures
Configuration II: 5,000 RPM

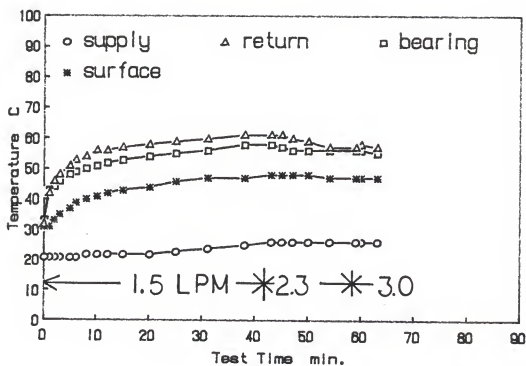


Figure 5.8 Transient Temperatures
Configuration II: 7,000 RPM

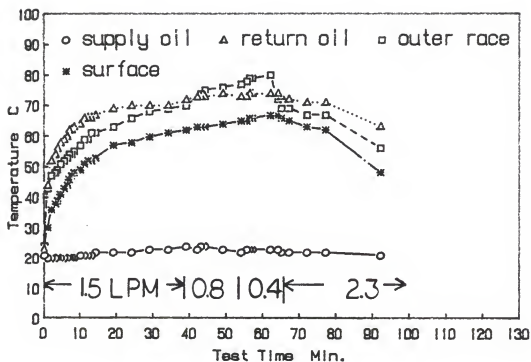


Figure 5.9 Transient Temperatures
Configuration II: 9,000 RPM

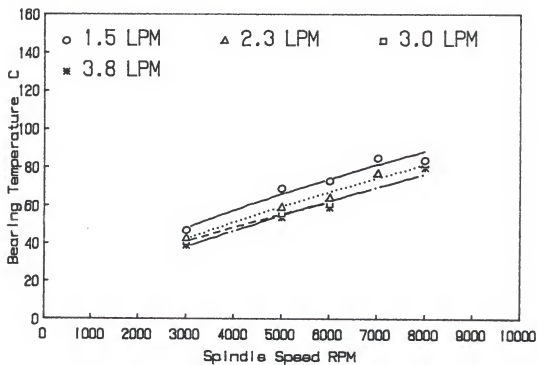


Figure 5.10 Bearing Temperature versus Spindle Speed
Configuration I

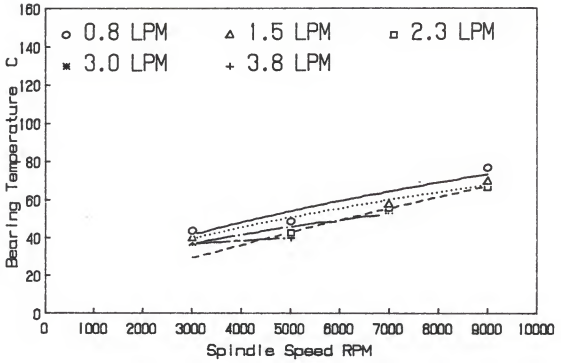


Figure 5.11 Bearing Temperature versus Spindle Speed
Configuration II

Table 5.1 Curve Fit Constants for Temperature Speed Relationships for Configuration I

Oil Flow Rate	a_1		a_2		R^2	
	a_1	a_2	a_1	a_2	R^2	R
0.8 lpm	0.33	0.62	0.96	0.98	0.98	0.99
1.5 lpm	0.21	0.66	0.98	0.99	1.00	1.00
2.3 lpm	0.39	0.58	0.98	0.99	1.00	1.00
3.0 lpm	0.13	0.70	0.98	0.99	1.00	1.00
3.8 lpm						

Table 5.2 Curve Fit Constants for Temperature Speed Relationships for Configuration II

Oil Flow Rate			2	
	a_1	a_2	R	R
0.8 lpm	0.67	0.52	0.91	0.95
1.5 lpm	0.78	0.49	0.98	0.99
2.3 lpm	0.01	0.76	1.00	1.00
3.0 lpm	1.28	0.42	0.90	0.95
3.8 lpm	0.90	0.15	1.00	1.00

Table 5.3 Curve Fit Constants for Temperature Oil
Flow Rate Relationships for Configuration I

Spindle Speed			2	
	a_3	a_4	R	R
3000 rpm	53.10	-0.24	0.99	0.99
5000 rpm	75.60	-0.26	0.87	0.93
6000 rpm	79.16	-0.23	0.97	0.98
7000 rpm	93.36	-0.23	1.00	1.00
8000 rpm	85.75	0.00	1.00	1.00
9000 rpm	-----	-----	-----	-----

Table 5.4 Curve Fit Constants for Temperature
Oil Flow Rate Relationships for Configuration II

Spindle Speed			2	
	a_3	a_4	R	R
3000 rpm	42.53	-0.11	0.97	0.99
5000 rpm	48.49	-0.12	0.84	0.92
6000 rpm	-----	-----	-----	-----
7000 rpm	59.81	-0.08	1.00	1.00
8000 rpm	-----	-----	-----	-----
9000 rpm	73.51	-0.10	0.96	0.98

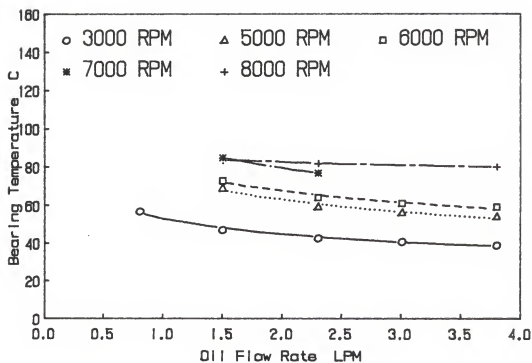


Figure 5.12 Bearing Temperature versus Oil Flow Rate
Configuration I

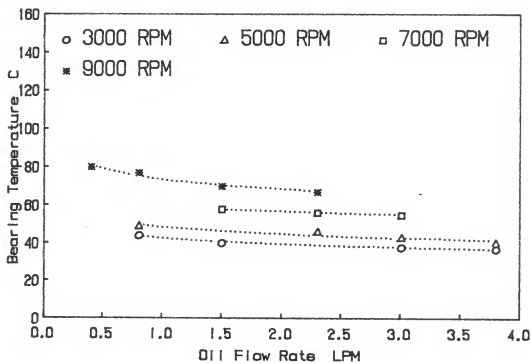


Figure 5.13 Bearing Temperature versus Oil Flow Rate
Configuration II

The approximation functions derived for the Bearing Temperature-Spindle Speed relationship, equation (5.1) with Table 5.1 and Table 5.2, and those for Bearing Temperature-Oil Flow Rate relationship, equation (5.2) with Table 5.3 and Table 5.4, match very well the experimental data. In Figure 5.12, the curve for 8,000 rpm does not follow the pattern of all the other constant speed curves since it never achieved steady state. During the test, the operating temperature exceeded 90°C before reaching steady state, therefore the oil flow rate was increased to avoid damaging the bearings.

Overall Temperature Equation

An approximation function for T_b , which includes both spindle speed and oil flow rate as variables, can now be derived using the method recommended by Johnson [35]. First, we obtain individual exponential relationships between each of the independent variables and the dependent variable, equations (5.1) and (5.2) with Tables 5.1 to Table 5.4. These equations are now combined by determining an average value for the exponent for each variable. The value of the constant for the overall equation is determined by substituting the experimental values of T_b , v and n , and solving for the constant. For Configuration I the resultant general approximation function is

$$T_B = 0.30 v^{-.24} n^{.64} \quad (^\circ\text{C}) \quad (5.3)$$

for the range: $1.5 \leq v \leq 3.8$ lpm

$$3,000 \leq n \leq 8,000 \text{ rpm}$$

For Configuration II, the derived general equation is

$$T_B = 0.97 v^{-0.10} n^{.47} \quad (^\circ\text{C}) \quad (5.4)$$

for the range: $1.5 \leq v \leq 3.8$ lpm

$$3,000 \leq n \leq 9,000 \text{ rpm}$$

The general approximation functions, equations (5.3) and (5.4) were used to generate Figures 5.14 and 5.15, for Configuration I and Figures 5.16 and 5.17, for Configuration II. Note that Figures 5.14 and 5.15 are similar to Figures 5.10 and 5.12, while Figures 5.16 and 5.17 are similar to Figures 5.11 and 5.13. As it can be seen from the figures, the general equations yield good approximations to the experimental data.

Steady State Temperatures: Comparison

Using the steady state operating temperature as a performance parameter, we can conclude from the experimental data that Configuration I should not be operated at spindle speeds above 6,000 rpm. Configuration II, on the other hand, performs such that operating temperatures at 9,000 rpm and 1.5 lpm are acceptable. For all combinations of oil flow rate and speed Configuration II operated at significantly lower temperatures than Configuration I. As it will be shown later, power losses for Configuration II were always lower than for Configuration I.

Spindle speed is the parameter with the strongest effect on bearing temperature. For Configuration I the effect was stronger than for Configuration II, due to the bearing geometry and number of rolling elements for each configuration. The effect of oil flow rate on bearing temperature was also stronger on Configuration I than for Configuration II, due to the inherent oil flow rates patterns for each bearing. In Configuration II, the oil flows efficiently through the bearing while in the CRB used in Configuration I, the bearing is splashed every which way leaving sections of the bearing without appropriate cooling [16]. Therefore, an increase in oil flow rate would be more significant for Configuration I, since the excess oil would now reach more of the bearing surface.

For both configurations the general approximations provide a good agreement with the experimental data. Therefore, these general equations (5.3) and (5.4) could be used to estimate the operating temperatures of these types of bearings within the test range.

It should be noted here that the curves describing the relationship between operating temperature and oil flow rate/spindle speed coincide in shape with those presented by other researchers in [21,22,23,24,25 and 26].

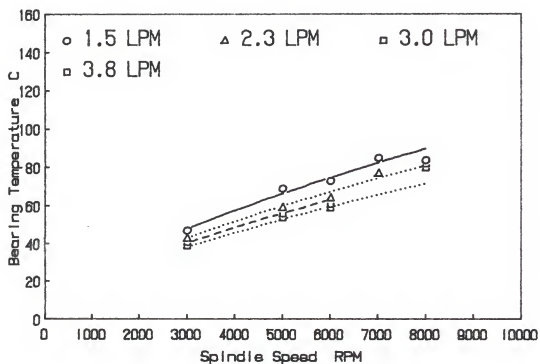


Figure 5.14 Bearing Temperature versus Spindle Speed
Configuration I (Overall)

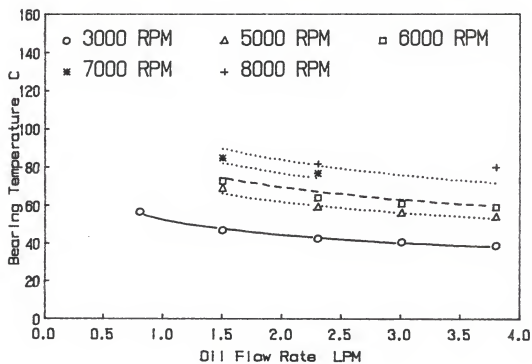


Figure 5.15 Bearing Temperature versus Oil Flow Rate
Configuration I (Overall)

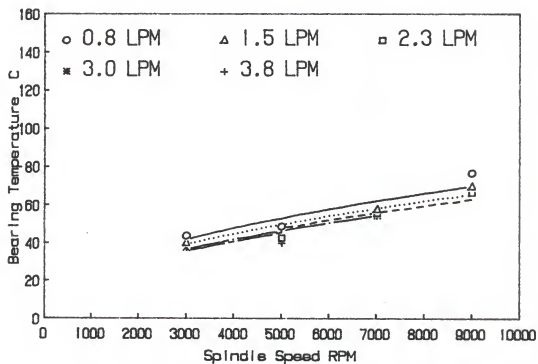


Figure 5.16 Bearing Temperature versus Spindle Speed
Configuration II (Overall)

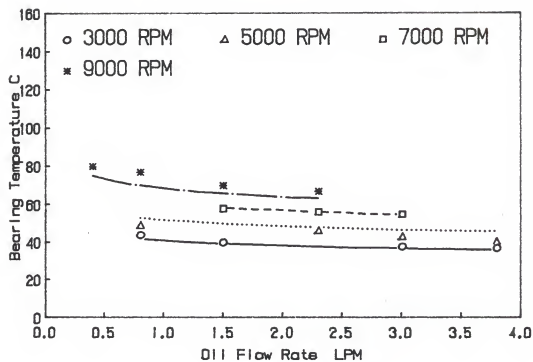


Figure 5.17 Bearing Temperature versus Oil Flow Rate
Configuration II (Overall)

Power Measurements

Three main types of power losses were identified from the test. There is the motor power losses P_{motor} , the mechanical power losses P_{me} , and the hydraulic power losses P_h . These power losses are described next. The identification of these power losses allows the comparison of different spindle configurations based on these losses. As it will be shown next, these losses are independent of the machine the spindle is installed on.

Motor Power Losses

The motor power losses P_{motor} are independent of the spindle mounted on the machine. They are produced by the inefficiencies of the D.C. motor and the mechanical losses in the intermediate shaft of the transmission system. To measure P_{motor} , the spindle is disengaged from the transmission by removing the belt between the intermediate shaft and the spindle. The motor is then accelerated to test speed, driving with it the intermediate shaft. The input power is recorded. The input power is in this case the motor power losses.

Mechanical Power Losses

The mechanical power losses, P_{me} , are produced by mechanical inefficiencies in the spindle configuration, such as the friction between rolling elements and rings, see

Friction in Bearings section in Chapter IV. To determine P_{me} , the spindle is driven to test speed without circulating oil. The input power is recorded. To calculate P_{me} , we subtract P_{motor} from the input power. The difference is the mechanical power losses. In equation form

$$P_{me} = P_{input} - P_{motor} \quad (\text{kW}) \quad (5.5)$$

where

P_{input} : Power imputed to the D.C. motor at any
set of operating conditions (kW)

P_{motor} : Motor power losses (kW)

P_{me} : Mechanical power losses (kW)

Hydraulic Power Losses

The hydraulic losses are caused by viscous friction on the rolling elements by the circulating oil. These losses will depend on the type of oil, the oil flow rate, spindle speed and bearing geometry. To measure the hydraulic power losses P_h , the spindle is driven to the test speed and oil flow rate. The input power is recorded. From the input power, the mechanical and motor power losses are subtracted. The difference is the hydraulic power losses P_h , for the given speed and oil flow rate.

Since P_{me} and P_h are characteristic for each configuration, they can be used as comparison parameters. For any configuration, the lower the power losses are the

more efficient is the configuration. The lower the power losses, the lower the cooling requirements, since power losses are the source of heat.

Configuration Power Losses

Mechanical Power Losses

The mechanical power losses for each configuration are shown in Figure 5.18. The equations which describe the mechanical power losses for each configuration are given next.

for Configuration I:

$$P_{me} = 8.6 \times 10^{-4} n \quad (\text{kW}) \quad (5.6)$$

for Configuration II:

$$P_{me} = 5.6 \times 10^{-4} n \quad (\text{kW}) \quad (5.7)$$

where n is spindle speed (rpm).

Since there is no oil being circulated, there is no dependence on oil flow rate. By comparing equations (5.6) and (5.7) it should be clear that the mechanical power losses for Configuration II are lower than for Configuration I. This is also shown in Figure 5.18.

Hydraulic Power Losses

The effect of spindle speed and oil flow rate on hydraulic power losses was investigated. The graphs which will be discussed next represent the power losses for each

configuration. The indicated flows are per bearing, not total oil flow rate. For Configuration I, the three bearings received the same oil flow rate at all times. For Configuration II, the HYDRA-RIB_{TM} was supplied with 3.8 lpm at all times. The flow through the High Speed bearing was varied from 0.8 lpm to 3.8 lpm. The graphs indicate the total hydraulic power loss for the configuration, not for individual bearings.

Hydraulic Power Losses Versus Spindle Speed

The effect of spindle speed, at constant oil flow rate, on hydraulic power losses is shown in Figure 5.19 and Figure 5.20, for Configuration I and Configuration II, respectively. As before, the experimental data is represented by the symbols and the curves represent approximation functions of the data. The approximation functions were derived as before, using a least square scheme. The approximation functions are of the form

$$P_h = a_5 n^{a_6} \quad (^\circ\text{C}) \quad (5.8)$$

where

P_h : Hydraulic Power Losses (kW)

n : Spindle Speed (rpm)

a_5 : Curve fit constant one (kW/rpm ^{a_6})

a_6 : Curve fit constant two

The value of a_5 and a_6 for each configuration, for constant oil flow rates, are tabulated in Table 5.5 and Table 5.6. The approximation functions indicate that hydraulic power losses increase as spindle speed increases. This is shown graphically in Figures 5.19 and 5.20. As it can be seen from the figures, at 8,000 rpm and 3.8 lpm the hydraulic power losses reach 14 kW for Configuration I. If the mechanical power losses are added, the total power losses amount to 20 kW. This is too large to be acceptable. Thus, Configuration I is not recommended for use at these high spindle speeds.

Figures 5.19 and 5.20 show that the power losses of Configuration II, are lower than those of Configuration I. The steady state hydraulic power losses for Configuration II at 9,000 rpm and 2.3 lpm are 9.7 kW. This is almost 2 kW lower than the power losses for Configuration I at 8,000 rpm and the same oil flow rate. In Configuration II the effect of speed on the hydraulic power losses is quite strong, as demonstrated in the figures.

Hydraulic Power Losses Versus Oil Flow Rate

The effect of oil flow rate on hydraulic power losses is shown in Figures 5.21 and 5.22 on Configuration I and Configuration II, respectively for constant spindle speed. The approximation functions were derived as before and are in the form

$$P_h = a_7 v^{a_8} \quad (^\circ\text{C}) \quad (5.9)$$

where

P_h : Hydraulic Power Losses (kW)

v : Oil Flow Rate (lpm)

a_7 : Curve fit constant one (kW/lpm ^{a_8})

a_8 : Curve fit constant two

The corresponding values for a_7 and a_8 are tabulated in Table 5.7 and Table 5.8 for constant spindle speed. For Configuration I and II, hydraulic power losses increase with oil flow rate. This is shown graphically in Figures 5.21 and 5.22. The figures also show that the power losses of Configuration II are lower than those of Configuration I. Configuration II is very insensitive to increases in oil flow rate, this is due to the pumping action inherent in tapered roller bearings, which facilitate the flow of oil through the bearings.

Overall Hydraulic Power Losses Equation

As before, a general approximation equation was derived which combines the effects of spindle speed and oil flow rate on hydraulic power losses. These general approximation equations are given as

for Configuration I:

$$P_h = 0.0000030 v^{0.82} n^{1.6} \quad (\text{kW}) \quad (5.10)$$

for $3,000 \leq n \leq 8,000$ rpm

for $1.5 \leq v \leq 3.8$ lpm

for Configuration II:

$$P_h = 0.000016 v^{0.15} n^{1.4} \quad (\text{kW}) \quad (5.11)$$

for $3,000 \leq n \leq 9,000$ rpm

for $0.80 \leq v \leq 3.8$ lpm

Using equations (5.10) Figures 5.23 and 5.24 were produced for Configuration I. While Figures 5.25 and 5.26 were generated using equation (5.11), for Configuration II. These figures are similar to Figures 5.18 to 5.22, for the respective configuration. As it can be seen from the figures, the agreement of the general equation for P_h is not as good as the general equation for T_B . The general approximation function tends to predict higher power losses than those actually measured.

Power Losses: Comparison

Configuration II operated at all times with lower power losses than Configuration I. The lower power losses for Configuration II are due in part to the fact that Configuration II is based on two bearings while Configuration I is based on three. The geometry of the bearing has the strongest effect on bearing temperature. The tapered bearings have an inherent pumping action which facilitates the flow of oil through the bearings, providing for better oil flow and heat removal. The bearings in Configuration I are double row bearings which receive the

lubrication between both rows, after the oil cools the outer ring. The oil is then splashed towards the sides and between the bearings. This causes the large increase in hydraulic power losses with little effect on bearing temperature.

By observing Figures 5.10, 5.12, 5.19 and 5.21, it can be concluded that an oil flow rate of around 2.3 lpm per bearing would provide the best combination of operating temperature and power losses for Configuration I. The reduction in operating temperature does not justify the increase in power losses which occur once the oil flow rate exceeds 2.3 lpm.

It should be noted here that the curves describing the relationship between power losses and oil flow rate/spindle speed coincide in shape with those presented by other researchers in [21,22,23,24,25 and 26].

For Configuration II, Figures 5.11, 5.13, 5.20 and 5.22 show that a minimum oil flow rate of 1.0 lpm to the High Speed bearing would provide the best combination of operating temperature and power losses. As Figures 5.14 and 5.15 illustrate, the decrease in temperature due to increased flow is insignificant after 1.0 lpm. Note that the Hydra-Rib_{TM} bearing is receiving a flow of 3.8 lpm at all times. The 1.0 lpm oil flow rate to the High Speed bearing should be provided as 0.5 lpm to the front and 0.5 lpm to the back through the cone-rib lubrication holes.

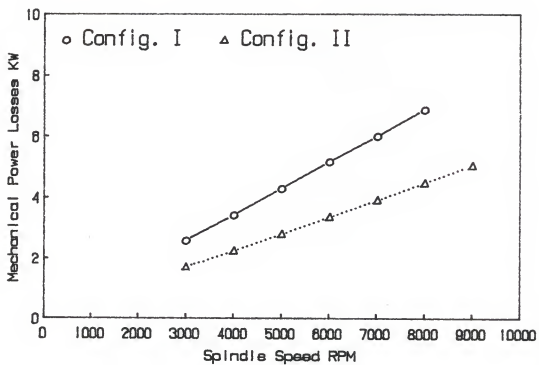


Figure 5.18 Mechanical Power Losses
Configurations I and II

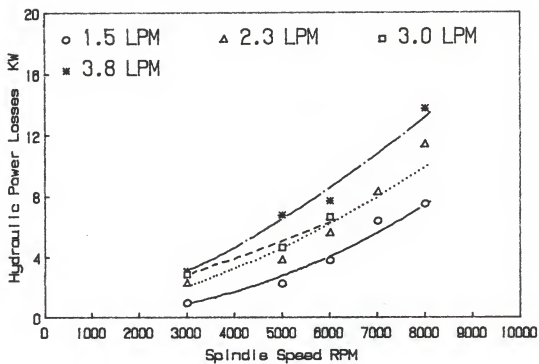


Figure 5.19 Hydraulic Power Losses versus Spindle Speed
Configuration I

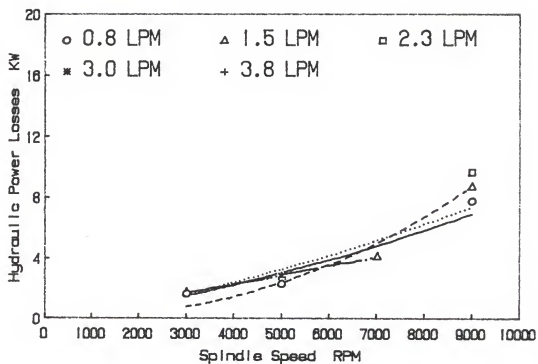


Figure 5.20 Hydraulic Power Losses versus Spindle Speed Configuration II

Table 5.5 Curve Fit Constants for Hydraulic Power Losses Speed Relationships for Configuration I

Oil Flow Rate	2		2	
	a5	a6	R	R
0.8 lpm	----	----	----	----
1.5 lpm	3.6 E-8	2.1	0.98	0.99
2.3 lpm	5.0 E-6	1.6	0.95	0.97
3.0 lpm	2.6 E-4	1.2	0.97	0.98
3.8 lpm	1.9 E-5	1.5	0.99	0.99

Table 5.6 Curve Fit Constants for Hydraulic
Power Losses Speed Relationships for Configuration II

Oil Flow Rate	a5	a6	R2	R
0.8 lpm	1.8 E-5	1.4	0.92	0.96
1.5 lpm	2.3 E-5	1.4	0.94	0.97
2.3 lpm	1.4 E-8	2.2	0.94	0.97
3.0 lpm	3.3 E-4	1.1	1.00	1.00
3.8 lpm	5.6 E-4	1.0	1.00	1.00

Table 5.7 Curve Fit Constants for Hydraulic
Power Losses Oil Flow Rate Relationships for Configuration I

Spindle Speed			2	
	a_z	a_s	R	R
3000 rpm	0.91	1.0	0.92	0.96
5000 rpm	1.4	1.1	0.99	1.00
6000 rpm	2.9	0.76	0.99	1.00
7000 rpm	5.0	0.60	1.00	1.00
8000 rpm	6.1	0.64	0.94	0.97
9000 rpm	----	----	----	----

Table 5.8 Curve Fit Constants for Hydraulic
Power Losses Oil Flow Rate Relationships
for Configuration II

Spindle Speed			2	
	a_7	a_8	R	R
3000 rpm	1.7	0.0	----	----
5000 rpm	2.4	0.13	0.93	0.96
6000 rpm	----	----	----	----
7000 rpm	4.2	0.0	0.00	0.00
8000 rpm	----	----	----	----
9000 rpm	8.2	0.18	0.99	0.99

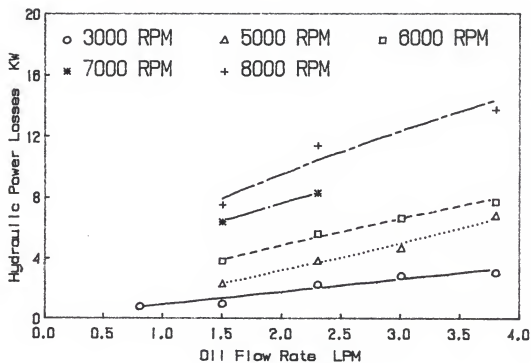


Figure 5.21 Hydraulic Power Losses versus Oil Flow Rate
Configuration I

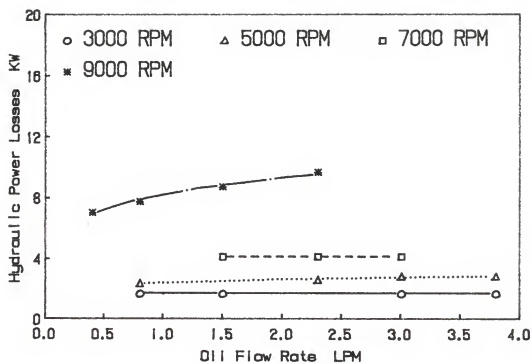


Figure 5.22 Hydraulic Power Losses versus Oil Flow Rate
Configuration II

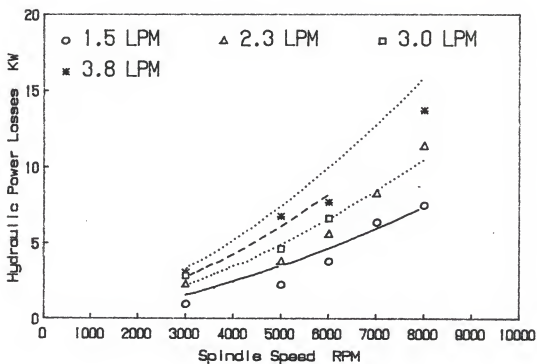


Figure 5.23 Hydraulic Power Losses versus Spindle Speed
Configuration I (Overall)

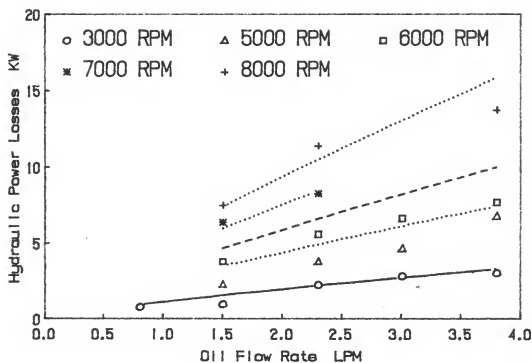


Figure 5.24 Hydraulic Power Losses versus Oil Flow Rate Configuration I (Overall)

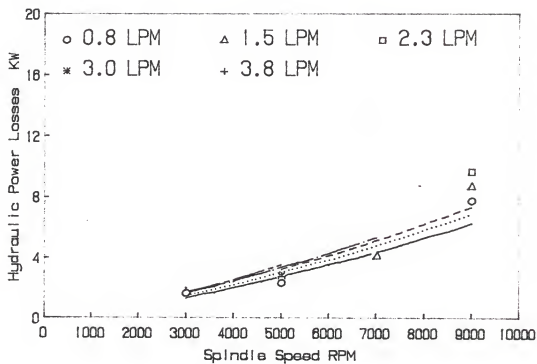


Figure 5.25 Hydraulic Power Losses versus Spindle Speed
Configuration II (Overall)

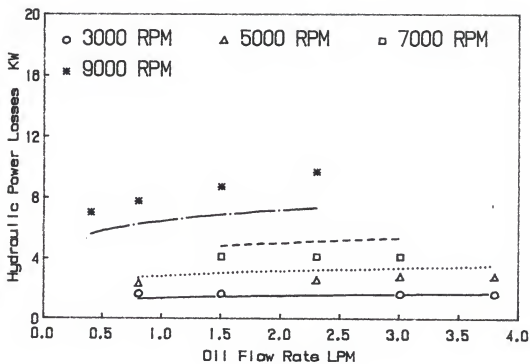


Figure 5.26 Hydraulic Power Losses versus Oil Flow Rate
Configuration II (Overall)

Bearing Loads

Externally Applied Load

The only externally applied load acting on the bearings during the investigation was induced by the tension of the driving belt on the spindle. The magnitude of the belt tension may be computed using equations provided by the belt manufacturer for each belt type and the dimension of the pulleys. Two different belt types were used, HABASITTM types A-5 (Configuration I) and S-5 (Configuration II). Both are rated to operate at 150 hp. and 10,000 rpm. The S-5 type belt is recommended to operate with less noise, while type A-5 is recommended for heavy shock loads. Both belts operated satisfactory under the idle test conditions. During the cutting test, the belts slipped when heavy cuts were attempted. Belt type S-5 was considerably quieter than type A-5.

The load on the bearings was computed from data and equations supplied by the belt manufacturer [31]. The manufacturer uses a combination of empirical factors based on the belt type, operating speed, and pulley diameter. For each one of these parameters, an empirical factor is provided in [31]. Using this publication [31], the belt tension load induced on Configuration I was computed at 8600 N while for Configuration II it was computed at 3600 N. As it was mentioned in Chapter IV, the individual bearing loads are on the lower bearing (NN3022 K) 3000 N, for the top bearing (NN3019 K) 12000 N and for the center bearing, the

Radiax, 400 N which is the weight of the spindle. For Configuration II, the load acting on the lower bearing, the High Speed Bearing, was estimated at 1600 N, while at the Hydra-Rib_{TM} the belt tension component was 5200 N and an axial component of 400 N due to the weight of the spindle.

Bearing Thermal Loads

The loads on the bearings were monitored using the PROMESS sensor. The PROMESS sensor provides a signal proportional to the load on the bearing elements. By monitoring this signal, the load on each bearing can be measured during the operation of the spindle. Therefore, the effects of speed and thermal expansions can be determined during transients and during steady state operation of the spindle.

The loads on Configuration I during one of the tests is shown in Figure 5.27. The increase in load due to differential thermal expansions can be clearly observed in the figure. As the test progresses, the thermally induced load increases, peaks and starts to decrease as the housing starts to expand. Finally, only the speed induced loads remain.

The PROMESS sensor control box for Configuration II was damaged so the controller for Configuration I was used instead. This provided a signal which increased with bearing load although not reliably enough to talk about

absolute load magnitude. The observed signal indicated that the constant preload mechanism worked effectively, avoiding drastic changes in load during transients.

The loads measured using the PROMESS sensor were relatively low for the configurations considering the type of bearing used. This leads into the conclusion that for the configurations investigated and the lubrications schemes used, thermal differential expansions are not a problem.

Performance of the Seals

As it was mentioned in Chapter III, each spindle configuration was provided with non-contact labyrinth seals. In general, the seals performed as designed when operated below 8,000 rpm. Once over this speed, oil mist was produced by small oil droplets which escaped the spindle housing. These oil droplets were then changed into mist by the spindle induced whirl surrounding the housing.

To avoid the mist, the oil flow rate could be reduced, or an air seal could be added to the labyrinth seal, or the oil could be removed from the housing more efficiently. At the high spindle speeds at which the mist occurs, reduction of the oil flow rate would produce an increase in bearing temperature, this is not acceptable.

An air seal was tested to improve the labyrinth seal in Configuration II. The air flow which was required to produce effective sealing action affected the lubricant flow through the high speed bearing. The air flowing into the

housing blocked the lubricant flow into the cone-rib contact causing the seizure of the bearing. Thus, the use of an air seal is not recommended for Configuration II.

To improve the removal of oil from the housing, rings could be provided to fling the oil away from the spindle and into grooves in the housing. From these grooves the oil could be sucked away avoiding the excess oil at the bottom of the spindle housing. For Configuration II, a more efficient oil distribution system could avoid the accumulation of oil at the bottom of the housing. If the ring which supplies oil to the Hydra-RibTM is redesigned such that instead of eight supply holes, only three are provided, the amount of oil getting to the lower end of the housing would be reduced. As the ring was originally designed, the oil is not forced high enough into the bearing, causing it to fall back down into the housing, eventually reaching the lower end.

Bearing Failures

Three bearings failed catastrophically during the investigation. For Configuration I, the Radiax bearing seized twice during the investigation. The failures occurred when the spindle was driven to 10,000 rpm. The other bearing failure occurred when an air seal was used in Configuration II. As the air seal was tested, the spindle was driven above 9,000 rpm, when the HS bearing failed.

During each of the test where a bearing failed, sufficient amounts of oil were circulated through the bearings. Bearing failures which are not caused by material fatigue are considered to be premature [9]. If the bearing is properly lubricated the bearing should not fail. Based on this fact the bearing failures are discussed next.

Radiax Bearing Failure

The Radiax bearing failed while Configuration I was driven to 10,000 rpm. The oil flow rate to the bearing at the time of the test was 3.8 lpm, which was considered sufficient based on previous test to 8,000 rpm. Once the failed bearing was removed from the spindle, it was inspected to determine the cause of the failure. The lower of the two rows of balls was intact and in excellent working conditions. The rolling elements in the upper row were, on the other hand, welded to the races and cage. The cage showed considerable deformation. The rollers exhibit considerable skidding and high temperature exposure. Comparing the bearing surface to the available literature on bearing failures [9,10,30], it was concluded that it was not a material fatigue induced failure. The skid marks indicated that the elements were not sufficiently preloaded. Also, when skidding occurs, the heat generation rates increase considerably so that the lubrication needs are greater. Thus, it was concluded that the preload acting on

the upper row was eliminated due to the high speed rotation, this produced the rolling elements to skid. Once the balls started skidding, the surface was damaged and the temperature increased to the point where the rolling elements welded themselves to the raceways. The bearing manufacturer (SKF) confirmed this conclusion upon inspection of the bearing.

To limit skidding at high speeds, the preload must be increased. To increase the preload, the separating ring between the two rows of balls must be ground. The increase in preload will be proportional to the reduction in distance between the rows. On the other hand, an increase in bearing load would increase the heat generation rates, as shown in Chapter IV. Larger heat generation rates would limit the operating speed to less than 8,000 rpm, since at that speed the operating temperature was already too high. Therefore, increasing the preload would eliminate skidding, but would increase the operating temperature above the safe limit.

To reduce the operating temperature, the lubrication system should be modified. Oil jets should be added to provide lubrication to the raceways directly instead of using exclusively the holes in the outer ring. It could be argued that the lubrication value of this scheme is minimal, since the oil would be splashed away from the balls. On the other hand, the objective of these jets is to provide cooling to the elements (balls, raceways and cage) not lubrication. From the test data, it was concluded that

increasing the oil flow rate above 2.3 lpm would not reduce the operating temperature significantly. Thus, 2.3 lpm could be provided by the current lubrication scheme and 1.5 lpm would be provided through the jets directly to the rolling elements. This should decrease the bearing temperature.

High Speed Bearing Failure

As mentioned above, the HS bearing in Configuration II failed during the test of an air seal. The spindle was operated in excess of 9,000 rpm and oil flow rates larger than 2.3 lpm. In previous tests, without the air seal, the bearing operated satisfactory at these conditions. Examination of the failed bearing indicated that the cone-rib contact area suffered from lubricant starvation. The rear end of the rollers was considerably damaged and the surface discoloration indicates that the temperature exceeded the recommended maximum.

At 10,000 rpm, the DN value for this bearing is one million. If the cone-rib contact is neglected and lubrication is not provided for this critical area, the usual DN limit is 0.5 million DN [21,22,23,25]. For the high speed tapered bearing used, 3.0 million DN has been experimentally obtained [26,27]. Thus, the bearing was receiving some cone-rib lubrication, but not enough to allow the bearing to exceed 1.0 million DN.

As it was mentioned before, the cause of the failure was that the pressurized air supplied to the seal affected the circulation of the oil to the cone-rib contact. Preliminary test showed that the spindle was capable of operating at spindle speeds in excess of 10,000 rpm. Thus, the air seal was eliminated as a solution for the oil mist problem.

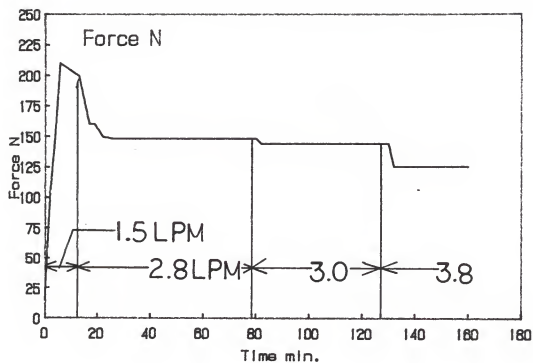


Figure 5.27 Bearing Thermal Loads

CHAPTER VI
BEARING LOADS

Load Deflection Relationships

The loads exerted on a bearing are transmitted through the contacts between the rolling elements and the raceways. The geometry of the rolling elements and their material properties determines how well the bearing can transmit the loads during operation. Heat generation and power losses are also a function of the contact geometry.

Let's consider a contact between a rolling element and a raceway. Hertz derived the equations for two bodies in contact in 1881, references [30,36]. The basic assumptions made by Hertz in his analysis were:

1. The materials are homogeneous and the yield strength is not exceeded.
2. No tangential forces are induced between the solids.
3. Contact is limited to a small portion of the surface.
4. The solids are at steady state.

For a steel ball/raceway contact the load deflection equation is given by

$$F = 2.14 \times 10^5 \delta^{* -2/3} \Sigma q^{-1/2} \delta^{3/2} \quad (N) \quad (6.1)$$

or

$$F = k_b \delta^{3/2} \quad (N) \quad (6.2)$$

where

F : normal force between rolling element and raceway (N)

δ : deformation (mm)

δ^* : dimensionless contact deformation, function of the curvature difference

$\Sigma \rho$: curvature sum (mm^{-1})

k_b : load deflection coefficient for a contact between a ball and one ring ($\text{N/mm}^{1.5}$)

The dimensionless deflection δ^* , is given by Harris, in a set of graphs, as functions of $F(\)$. These graphs were computed using elliptical integrals. Hamrock and Dowson [36] provide a simplified method for the computation of the elliptical integrals and the contact deflection. The method was developed by Brewe and Hamrock in 1977 by using least square regression analysis of the relationship between the elliptical integrals and the curvature difference. Using this method, the deflection is given by

$$\delta = \left(\frac{F}{K} \right)^{2/3} \quad (\text{mm}) \quad (6.3)$$

where

$$K = \pi k' E' \left(\frac{RE}{4.5F^3} \right)^{1/2} \quad (6.4)$$

$$k' = 1.0339 \left(\frac{R_y}{R_x} \right)^{0.6360} \quad (6.5)$$

$$E' = \frac{2}{\left(\frac{1-\mu_a^2}{E_a} \right) + \left(\frac{1-\mu_b^2}{E_b} \right)} \quad (6.6)$$

$$E = 1.0003 + \frac{0.5968}{R_y} R_x \quad (6.7)$$

$$F = 1.5277 + 0.6023 \ln(R_y/R_x) \quad (6.8)$$

$$R_y = \frac{d_m + D \cos \alpha}{2d_m} \quad (6.9)$$

$$R_x = \frac{fD}{2f-1} \quad (6.10)$$

f_o : conformity, r/D , for outer race

f_i : conformity, r/D , for inner race

E_a : Young's Modulus for body a (N/mm^2)

E_b : Young's Modulus for body b (N/mm^2)

For a steel roller/raceway contacts, the load deflection equation is given as

$$F = 7.86 \times 10^4 l^{8/9} \delta^{10/9} \quad (N) \quad (6.11)$$

or

$$F = k_x \delta^{10/9} \quad (N) \quad (6.12)$$

where

l : roller effective length (mm)

k_x : load deflection coefficient for a contact between a roller and one ring ($N/mm^{10/9}$)

The total approach of the two raceways for any rolling element under load is the sum of the approaches at each element/raceway contact, or

$$\delta_{\text{total}} = \delta_{\text{inner}} + \delta_{\text{outer}} \quad (\text{mm}) \quad (6.13)$$

where

δ_{total} : total approach for any element considering both raceways (mm)

δ_{inner} : approach at the inner race contact (mm)

δ_{outer} : approach at the outer race contact (mm)

In the case of ball bearings, δ_{inner} and δ_{outer} are different from each other, due to the change in curvature at each raceway. If we express the deflection in equation (6.2) and substitute into equation (6.13)

$$\left(\frac{F}{K_B} \right)^{2/3} = \left(\frac{F}{k_b} \right)^{2/3} + \left(\frac{F}{k_b} \right)^{2/3} \quad (6.14)$$

where

K_B : load deflection constant for a ball considering both contacts ($\text{N/mm}^{1.5}$)

Eliminating the force F from equation (6.14) and expressing K_b :

$$K_B = \left(\frac{1}{\left(\frac{1}{k_b} \right)^{2/3}_{\text{inner}} + \left(\frac{1}{k_b} \right)^{2/3}_{\text{outer}}} \right)^{3/2} \quad (6.15)$$

Then the load deflection equation for a ball considering both contacts is given by equation (6.16) below

$$F = K_B \delta^{3/2} \quad (\text{N}) \quad (6.16)$$

For symmetrical roller bearings, the deflection at the inner and outer races will be the same, then equation (6.13) can be rewritten as

$$\delta_R = 2\delta \quad (N) \quad (6.17)$$

where

δ_R : total approach for each roller considering both contacts (mm)

Expressing δ in equation (6.12) and substituting into equation (6.17)

$$\left(\frac{F}{K_R}\right)^{9/10} = 2 \left(\frac{F}{k_x}\right)^{9/10} \quad (\text{mm}) \quad (6.18)$$

where

K_R : load deflection coefficient for a roller considering both approaches ($N/\text{mm}^{10/9}$)

Eliminating the force F from equation (6.18) and solving for K_R

$$K_R = \left(\frac{1}{2^{10/9}(1/k_x)^{9/10}}\right)^{10/9} = 0.463k_x \quad (N/\text{mm}^{10/9}) \quad (6.19)$$

The load deflection relation for rollers is then

$$F = K_R \delta^{10/9} \quad (N) \quad (6.20)$$

Therefore, we can rewrite a general load deflection equation for either balls or roller elements

$$F = K \delta^q \quad (N) \quad (6.21)$$

where

q : exponent characteristic of the rolling element
 $= 3/2$ (1.5) for balls
 $= 10/9$ (1.11) for rollers

Now that the load deflection characteristics for a single rolling element is known, the load deflection behavior of a complete bearing will be consider.

Radial Loads

Let's consider now radial loads on a radial bearing. Radial bearings are those bearings design to support only radial loads. Two of the most common radial bearings are deep groove ball bearings and cylindrical roller bearings. Analyzing a radial bearing as the one in Figure 6.1, the deflection at the θ_j^{th} angular position is given by

$$\delta_{e_j} = \delta_r \cos \theta_j - 0.5 P_d \quad (\text{mm}) \quad (6.22)$$

where

$$j = 1, 2, \dots, z$$

z : number of rolling elements

P_d : diametral clearance (mm)

The force at angular position θ_j is given by

$$F_{e_j} = K \delta_{e_j}^q \quad (\text{N}) \quad (6.23)$$

For equilibrium to be satisfied, the magnitude of the sum of the internal radial components must add to the same magnitude as the applied external radial load or

$$F_r = \sum_{j=1}^z F_{e_j} \cos \theta_j \quad (\text{N}) \quad (6.24)$$

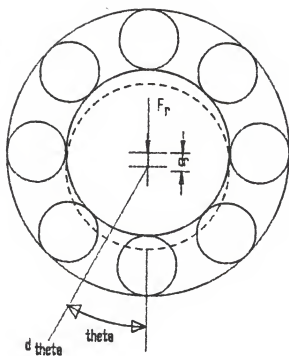


Figure 6.1 Loads on a Radial Bearing

Substituting equation (6.23) into equation (6.24)

$$F_x = \sum_{j=1}^Z K \delta_{\theta j}^{\alpha} \cos \theta_j \quad (N) \quad (6.25)$$

Substituting equation (6.22) into equation (6.25)

$$F_x = \sum_{j=1}^Z K (\delta_x \cos \theta_j - 0.5 P_d)^{\alpha} \cos \theta_j \quad (N) \quad (6.26)$$

with the constraint that if $\delta_{\theta j} < 0$, then $F_{\theta j} = 0$.

For any given external radial load F_x , equation (6.26) must be solved for δ_x . With δ_x known, the deflection and load for each rolling element can be computed using equation (6.23). Rewriting equation (6.26) such that we can solve for those values of δ_x which will make it equal to zero

$$F_x - \sum K (\delta_x \cos \theta_j - 0.5 P_d)^{\alpha} \cos \theta_j = 0 \quad (6.27)$$

To solve for δ_x , the Newton-Raphson method for determining the roots of an equation is used. The method is based on equation (6.28).

$$p_{m+1} = p_m + \frac{f(p_m)}{f'(p_m)} \quad m=1,2,\dots \quad (6.28)$$

where

p_m : m^{th} iteration of the variable p

$f(p_m)$: function evaluated at the m^{th} iteration

$f'(p_m)$: derivative of the function

In the case under consideration, equation (6.27) is our f function while δ_x is our p variable. The derivative of equation (6.27) with respect to δ_x is given by equation (6.29)

$$f'(p_m) = \sum_{j=1}^Z K (\delta_x \cos \theta_j - 0.5 P_d)^{\alpha-1} \cos^2 \theta_j \quad (6.29)$$

The equation to be used to solve by iteration for δ_x is

$$\delta_x(m+1) = \delta_x(m) + A/B \quad (\text{mm}) \quad (6.30)$$

where

$$A: (F_x - \sum_{j=1}^Z K(\delta_x \cos \theta_j - 0.5 P_d)^{\alpha} \cos \theta_j)$$

$$B: \sum_{j=1}^Z K(\delta_x \cos \theta_j - 0.5 P_d)^{\alpha-1} \cos^2 \theta_j$$

A list of the computer program developed to determine δ_x is included in the Appendix. Several example calculations from reference [30] were performed and compared. The match was good.

Axial Loads

Considering a TRB under centric axial load, each roller contributes equally to support the load. The axial load F_a on a roller is given by

$$F_a = \frac{F}{Z \sin \alpha} \quad (N) \quad (6.31)$$

where α is the contact angle measured in degrees.

For static equilibrium to exist

$$F_{1x} - F_{fx} - F_{ox} = 0 \quad (6.32)$$

$$F_{1a} + F_{fa} - F_{oa} = 0 \quad (6.33)$$

For a single tapered roller, the forces acting on it are shown in Figure 6.2.

The load on the outer race F_o and the load on the inner race F_i are given by

$$F_o = F_{i\alpha} \frac{\cos \alpha_\varepsilon + \tan \alpha_i \sin \alpha_\varepsilon}{\sin(\alpha_o + \alpha_\varepsilon)} \quad (N) \quad (6.34)$$

$$F_i = \frac{F_{i\alpha}}{\sin \alpha_i} \quad (N) \quad (6.35)$$

Once the loads F_o and F_i are determined, the deflection at the roller can be determined as

$$\delta_o = \left(\frac{F_o}{k} \right)^{2/10} \quad (mm) \quad (6.36)$$

$$\delta_i = \left(\frac{F_i}{k} \right)^{2/10} \quad (mm) \quad (6.37)$$

where

δ_o : normal approach at the outer race (mm)

δ_i : normal approach at the inner race (mm)

The axial deflection at the outer $\delta_{o\alpha}$ and inner $\delta_{i\alpha}$ contact are given by equations (6.38) and (6.39) respectively

$$\delta_{o\alpha} = \frac{\delta_o}{\sin \alpha_i} \quad (mm) \quad (6.38)$$

$$\delta_{i\alpha} = \frac{\delta_i}{\sin \alpha_i} \quad (mm) \quad (6.39)$$

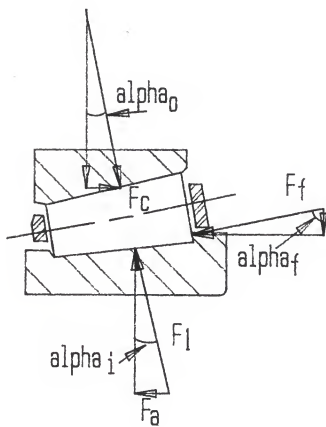


Figure 6.2 Axial Loads on a Tapered Roller

The total axial deflection $\delta_{a\text{total}}$ produced by the external axial force is then the sum of δ_{oa} and δ_{ia} .

$$\delta_{a\text{total}} = \delta_{oa} + \delta_{ia} \quad (\text{mm}) \quad (6.40)$$

The induced radial deflection at the outer δ_{or} and inner δ_{ir} contact are given by equations (6.41) and (6.42) respectively

$$\delta_{or} = \delta_o / \cos \alpha_o \quad (\text{mm}) \quad (6.41)$$

$$\delta_{ir} = \delta_i \cos \alpha_i \quad (\text{mm}) \quad (6.42)$$

The total radial deflection $\delta_{r\text{total}}$ produced by the external axial force is then the sum of δ_{or} and δ_{ir} .

$$\delta_{r\text{total}} = \delta_{or} + \delta_{ir} \quad (\text{mm}) \quad (6.43)$$

Combined Loading

If an ACBB or a TRB is under combined radial and axial loading, without diametral clearance, both races will remain parallel, as shown by Figure 6.3. There will be a displacement in the axial direction δ_a and a displacement in the radial direction δ_r . The total approach of the raceways is

$$\delta_{e_j} = \delta_a \sin \theta_j + \delta_r \cos \theta_j \cos \alpha - 0.5 P_a \quad (\text{mm}) \quad (6.44)$$

As before, the load deflection relationship for any element was given by equation (6.23) as

$$F_{e_j} = K \delta_{e_j}^q \quad (\text{N}) \quad (6.23)$$

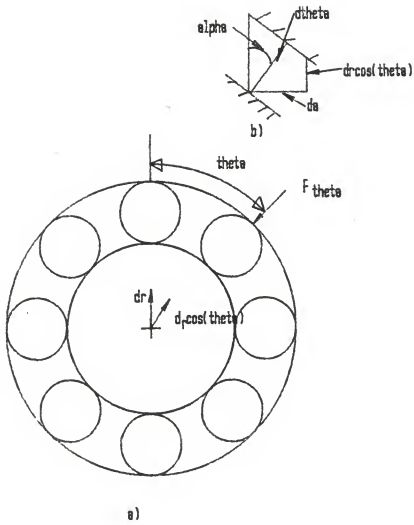


Figure 6.3 Combined Load Geometry
a) Geometry and Nomenclature;
b) Deflections.

Substituting equation (6.44) into equation (6.23) we obtain the force for the element at the θ_j position

$$F_{\theta_j} = K\delta_{\theta_j}^{\alpha} = K(\delta_a \sin\theta_j + \delta_r \cos\theta_j \cos\alpha - 1/2P_d)^{\alpha} \quad (6.45)$$

The axial and the radial components of the force F_{θ_j} are

$$F_{\theta_j \text{ axial}} = F_{\theta_j} \sin\alpha \quad (N) \quad (6.46)$$

$$F_{\theta_j \text{ radial}} = F_{\theta_j} \cos\theta \cos\alpha \quad (N) \quad (6.47)$$

with the constraint that if $\delta_{\theta_j} < 0$, then $F_{\theta_j} = 0$.

For equilibrium to exist, the sum of the magnitudes of the element forces, axially and radially, must equal the magnitude of the external axial and radial loads.

$$F_a = \sum_{j=1}^Z F_{\theta_j \text{ axial}} = \sum_{j=1}^Z F_{\theta_j} \sin\alpha \quad (N) \quad (6.48)$$

$$F_r = \sum_{j=1}^Z F_{\theta_j \text{ radial}} = \sum_{j=1}^Z F_{\theta_j} \cos\theta \cos\alpha \quad (N) \quad (6.49)$$

where

F_a : is the applied external axial load (N)

F_r : is the applied external radial load (N)

Substituting equation (6.45) into equations (6.48) and (6.49) we obtain the equations needed to determine δ_a and δ_r .

$$F_a = \sum_{j=1}^Z K(\delta_a \sin\theta_j + \delta_r \cos\theta_j \cos\alpha - 0.5 P_d)^{\alpha} \sin\alpha \quad (6.50)$$

$$F_r = \sum_{j=1}^Z K(\delta_a \sin\theta_j + \delta_r \cos\theta_j \cos\alpha - 0.5 P_d)^{\alpha} \cos\alpha \cos\theta_j \quad (6.51)$$

For any given set of external loads, the above two equations (6.50) and (6.51) must be solved for δ_a and δ_r . Using the Newton-Raphson method discussed above, both δ_a and δ_r can be determined. In this case, we have two p variables and two f functions. The functions are obtained from equations (6.50) and (6.51).

$$f_1(p_m) = F_a - \sum_{j=1}^Z K(\delta_a \sin \theta_j + \delta_r \cos \theta_j \cos \alpha - 0.5 P_a)^{\alpha} \sin \alpha \quad (6.52)$$

$$f_2(p_m) = F_r - \sum_{j=1}^Z K(\delta_a \sin \theta_j + \delta_r \cos \theta_j \cos \alpha - 0.5 P_a)^{\alpha} \cos \alpha \cos \theta_j \quad (6.53)$$

The derivatives of these equations, f_1 with respect to δ_a and f_2 with respect to δ_r are

$$f'_1(p_m) = \sum_{j=1}^Z K (\delta_a \sin \theta_j + \delta_r \cos \theta_j \cos \alpha - 0.5 P_a)^{\alpha-1} \sin \alpha \sin \theta_j \quad (6.54)$$

$$f'_2(p_m) = \sum_{j=1}^Z K (\delta_a \sin \theta_j + \delta_r \cos \theta_j \cos \alpha - 0.5 P_a)^{\alpha-1} (\cos \alpha \cos \theta_j)^2 \quad (6.55)$$

The equations used to determine the deflections are then

$$\delta_a(m+1) = \delta_a(m) + f_1(\delta_a(m))/f'_1(\delta_a(m)) \quad m=0,1,2,\dots \quad (6.56)$$

$$\delta_r(m+1) = \delta_r(m) + f_2(\delta_r(m))/f'_2(\delta_r(m)) \quad m=0,1,2,\dots \quad (6.57)$$

In the Appendix, there is also a list of the program developed for computing the axial and radial deflection for a given set of external loads. Again, the results of the program were compared with examples from reference [30], the agreement was good.

Bearing Life Calculation

The life of the bearing can be estimated using the standard equation [9,33]

$$\frac{L}{L_{10}} = \left(\frac{C}{P} \right)^p \quad 10^6 \text{ revolutions} \quad (6.58)$$

where

C: dynamic load rating of the bearing (N)

L_{10} : Rated Life (rev.)

P: equivalent load on the bearing (N)

p: 3 for ball bearings

10/3 for roller bearings

The dynamic load rating can be obtain from the manufacturers catalog [9,33] or can be computed as

$$C = f_c (\cos \alpha_o)^{0.7} z^{2/3} D^{1.8} \quad \text{for } D \leq 25.4 \quad (6.59)$$

$$C = f_c (\cos \alpha_o)^{0.7} z^{2/3} 3.647 D^{1.4} \quad \text{for } D \geq 25.4 \quad (6.60)$$

where

$$f_c = D \cos \alpha_o / d_m$$

i: number of rows

z: number of elements

D: rolling element diameter (mm)

For radial roller bearings, the equation to use is

$$C = f_c (i l_{eff} \cos \alpha_o)^{7/9} z^{3/4} D^{29/27} \quad (N) \quad (6.61)$$

where

l_{eff} : effective roller length (mm)

For thrust roller bearings

$$C = f_c (l_{eff} \cos \alpha_o)^{7/9} \tan \alpha_o z^{3/4} D^{29/27} \quad (N) \quad (6.62)$$

The load on the rolling element is given by the following equations, first for radial roller bearings

$$P = 0.401 F_m z \cos \alpha_o \quad (N) \quad (6.63)$$

for thrust bearings, ball or rollers

$$P = F_m z \sin \alpha_o \quad (N) \quad (6.64)$$

where

F_m : average element load

$$F_m = \sum_{i=1}^z \frac{F_i}{z} \quad (N) \quad (6.65)$$

Bearing Preload

As it was mentioned before, bearings in machine tools are installed with preload. Preload refers to the use of an initial or a mounting load on the bearings to improve its load deflection behavior. Preload eliminates the diametral clearance of the bearings, so all elements are in contact with both races. When a load is applied, all the elements participate in the support of the load, the individual rolling element load is then decreased, therefore, the deflection is reduced.

Radial bearings are preloaded by producing a radial interference. This interference is usually achieved by driving the bearing onto a tapered shaft. The more the bearing is driven onto the taper, the larger the

interference will be. The effect of preload on the load deflection behavior is shown in Figure 6.4. The slope of the load deflection curve is defined as the stiffness of the bearing.

In the case of the ACBB or TRB, one of the races is displaced axially more than the other, loading the elements with a force proportional to the relative displacement and contact angle.

Preloading Methods

In Figure 6.5, a and c, two different spindle bearing configurations are shown. In Figure 6.5a, both the inner and outer rings of a pair of bearings are kept at a constant distance from each other. A preload is obtained by means of an adjusting nut, which maintains a constant distance between both bearings. This is denoted as CASE 1. In Figure 6.5b, the direction of the azimuth angle θ for each bearing is indicated. Also in Figure 6.5b, the positive direction of the forces and deflections at each bearing is illustrated. The figure also illustrates the use of a spring (or a hydraulic cylinder) to provide a constant preload on the bearings. Let this be CASE 2.

Case 1: Variable Preload

In Figure 6.5a, the variable preload configuration is shown. The nut maintains a constant distance between the bearings. Thus, the axial deflection at each bearing should be of the same magnitude and in the same direction.

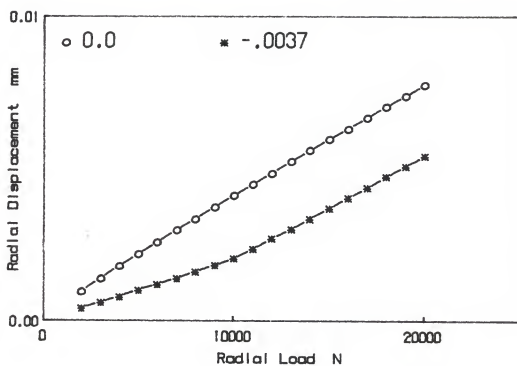


Figure 6.4 Effect of Preload on Bearing Stiffness

$$\delta_{xA} = -\delta_{xB} = \delta_x \quad (\text{mm}) \quad (6.66)$$

where

δ_{xA} : axial deflection of bearing A (mm)

δ_{xB} : axial deflection of bearing B (mm)

δ_x : axial deflection of the system (mm)

The axial force, acting on each bearing, is the combination of the preload and the resulting component of the external applied radial load. The axial load on each bearing, is of the same magnitude, but opposite direction.

$$F_{xA} = F_{xB} \quad (\text{N}) \quad (6.67)$$

where

F_{xA} : axial reaction on bearing A (N)

F_{xB} : axial reaction on bearing B (N)

The external radial load acts on each bearing in proportion to the distance from the bearing to the point of application.

$$F_{xA} = F_z \frac{(a+b)}{b} \quad (\text{N}) \quad (6.68)$$

$$F_{xB} = -F_z \frac{a}{b} \quad (\text{N}) \quad (6.69)$$

where

F_{xA} : radial load on bearing A (N)

F_{xB} : radial load on bearing B (N)

F_z : external radial load (N)

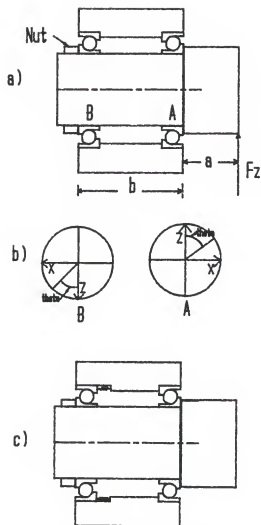


Figure 6.5 Preloading Methods

- a) Variable Preload Configuration;
- b) Definition of Positive Sense For Each Configuration;
- c) Constant Preload Configuration.

Using equation (6.25), the axial reactions F_{xA} and F_{xB} can be expressed as

$$F_{xA} = \sum_{j=1}^Z F_{\theta jA} \sin \alpha_A \quad (N) \quad (6.70)$$

$$F_{xB} = \sum_{j=1}^Z F_{\theta jB} \sin \alpha_B \quad (N) \quad (6.71)$$

The radial loads on the bearings, given equations (6.67) and (6.68) can be expressed using equation (6.48) as

$$F_{zA} = \sum_{j=1}^Z F_{\theta jA} \cos \alpha_A \cos \theta j_A = F_z (a+b)/b \quad (N) \quad (6.72)$$

$$F_{zB} = \sum_{j=1}^Z F_{\theta jB} \cos \alpha_B \cos \theta j_B = -F_z (a/b) \quad (N) \quad (6.73)$$

Equations (6.70) to (6.73) can be rewritten in terms of the external loads F_{xA} , F_{xB} , and F_z . The elements forces can be substituted for their deflections using equations (112) and (134), such that we are left with a system of three equations and three unknowns. The three unknown variables are δ_x , δ_{zA} and δ_{zB} . The three equations are

$$\sum_{j=1}^Z K(\delta_{zA} \cos \alpha \theta j_A \cos \alpha_A + \delta_x \sin \alpha_A + .5P_{dA})^{\alpha} \sin \alpha_A - \sum_{j=1}^Z EK(\delta_{zB} \cos \alpha \theta j_B \cos \alpha_B + \delta_x \sin \alpha_B + .5P_{dB})^{\alpha} \sin \alpha_B = 0 \quad (6.74)$$

$$\sum_{j=1}^Z EK(\delta_{zA} \cos \alpha \theta j_A \cos \alpha_A + \delta_x \sin \alpha_A + .5P_{dA})^{\alpha} \cos \alpha_A \cos \theta j_A = F_z (a+b)/b \quad (6.75)$$

$$\sum_{j=1}^Z EK(\delta_{zB} \cos \alpha \theta j_B \cos \alpha_B + \delta_x \sin \alpha_B + .5P_{dB})^{\alpha} \cos \alpha_B \cos \theta j_B = -F_z (a/b) \quad (6.76)$$

Equations (6.74), (6.75) and (6.76) are solved by using the Newton-Raphson method described before. The summation terms are computed first, then subtracted from the resultant force. The difference is divided by the corresponding derivative. The result is then added to the variable being iterated.

The Configuration I spindle is of the variable preload type, since the preload is applied during installation and there is no mechanism to maintain that preload. Configuration I differs from the one described above in that the radial support is produced exclusively by the radial bearings, while the Radiax supports all the axial loads.

To determine the deflections of each bearing, the load in the bearings must be determined. For the radial bearings, it is the case of a beam supported at two points under concentrated load. The reaction at each bearing is computed as above. The deflection at the radial bearing is computed using the technique discussed in section on radial loads, taking into consideration that the bearings now have two rows of elements.

The external axial load is entirely supported by the Radiax bearing. To determine the deflections, the double row bearing is treated as two, face to face, ACBBs under centric axial load.

Case 2: Constant Preload

In the spindle configuration shown in Figure 6.5c a constant preloading force is maintained on both bearings. Therefore, the axial force acting on each bearing is known at all times. The axial deflection will be different at each bearing. The axial forces at each bearing are equal to the preloading force.

$$F_{xA} = F_{xB} = F_D \quad (N) \quad (6.77)$$

In this configuration, if the sum of the axial components at one bearing, exceeds the external opposite axial load, the bearing will start moving in the direction of the axial component of the radial load. At this point, axial deflections become so large that further computation of the radial deflections is extremely difficult. As in Case 1, the radial force at each bearing is proportional to the external radial load.

$$F_{zA} = F_z \frac{a+b}{b} \quad (N) \quad (6.68)$$

$$F_{zB} = -F_z \frac{a}{b} \quad (N) \quad (6.69)$$

As before the following equations hold

$$F_{xA} = \sum_{j=1}^Z F_{\Theta jA} \sin \alpha_A \quad (N) \quad (6.70)$$

$$F_{xB} = \sum_{j=1}^Z F_{\Theta jB} \sin \alpha_B \quad (N) \quad (6.71)$$

Z

$$F_{zA} = \sum_{j=1}^Z F_{\theta jA} \cos \alpha_A \cos \theta_{jA} = F_z(a+b)/b \quad (N) \quad (6.72)$$

$$F_{zB} = \sum_{j=1}^Z F_{\theta jB} \cos \alpha_B \cos \theta_{jB} = -F_z(a/b) \quad (N) \quad (6.73)$$

Substituting for the deflections we obtain

$$F_{xA} = \sum_{j=1}^Z K(\delta_{zA} \cos \alpha_{\theta jA} \cos \alpha_A + \delta_{xA} \sin \alpha_A + 0.5 P_{dA})^\alpha \sin \alpha_A = F_P \quad (6.78)$$

$$F_{xB} = \sum_{j=1}^Z K(\delta_{zB} \cos \alpha_{\theta jB} \cos \alpha_B + \delta_{xB} \sin \alpha_B + 0.5 P_{dB})^\alpha \sin \alpha_B = F_P \quad (6.79)$$

$$\sum_{j=1}^Z K(\delta_{zA} \cos \alpha_{\theta jA} \cos \alpha_A + \delta_{xA} \sin \alpha_A + 0.5 P_{dA})^\alpha \cos \alpha_A \cos \theta_{jA} = F_z(a+b)/b \quad (N) \quad (6.80)$$

$$\sum_{j=1}^Z K(\delta_{zB} \cos \alpha_{\theta jB} \cos \alpha_B + \delta_{xB} \sin \alpha_B + 0.5 P_{dB})^\alpha \cos \alpha_B \cos \theta_{jB} = -F_z(a/b) \quad (N) \quad (6.81)$$

It is more convenient to rewrite equations (6.78) to (6.81) as

$$\sum_{j=1}^Z K(\delta_{zA} \cos \alpha_{\theta jA} \cos \alpha_A + \delta_{xA} \sin \alpha_A + 0.5 P_{dA})^\alpha \sin \alpha_A - F_P = 0 \quad (6.82)$$

$$\sum_{j=1}^Z K(\delta_{zB} \cos \alpha_{\theta jB} \cos \alpha_B + \delta_{xB} \sin \alpha_B + 0.5 P_{dB})^\alpha \sin \alpha_B - F_P = 0 \quad (6.83)$$

$$\sum_{j=1}^Z EK(\delta_{xA} \cos \theta_j \cos \alpha_A + \delta_{xB} \sin \alpha_A + 0.5 P_{dA})^{\alpha} \cos \alpha_A \cos \theta_j - F_z (a+b)/b = 0 \quad (6.84)$$

$$\sum_{j=1}^Z EK(\delta_{xB} \cos \theta_j \cos \alpha_B + \delta_{xA} \sin \alpha_B + 0.5 P_{dB})^{\alpha} \cos \alpha_B \cos \theta_j + F_z (a/b) = 0 \quad (6.85)$$

In Case 2, there are four equations and four unknowns. The four unknowns are δ_{xA} , δ_{xB} , δ_{zA} , and δ_{zB} . Again we use the Newton-Raphson method on equations (6.82) to (6.85) to solve for the deflections, given a set of external loads. The derivative of each equation with respect to the corresponding deflection is given in equations (6.86) to (6.87)

$$\frac{D(6.82)}{\delta_{xA}} = \sum_{j=1}^Z K q (\delta_{xA} \cos \theta_j \cos \alpha_A + \delta_{xB} \sin \alpha_A + 0.5 P_{dA})^{\alpha-1} \sin^2 \alpha_A \quad (6.86)$$

$$\frac{D(6.83)}{\delta_{xB}} = \sum_{j=1}^Z K q (\delta_{xB} \cos \theta_j \cos \alpha_B + \delta_{xA} \sin \alpha_B + 0.5 P_{dB})^{\alpha-1} \sin^2 \alpha_B \quad (6.87)$$

$$\frac{D(6.84)}{\delta_{zA}} = \sum_{j=1}^Z K q (\delta_{xA} \cos \theta_j \cos \alpha_A + \delta_{xB} \sin \alpha_A + 0.5 P_{dA})^{\alpha-1} (\cos \alpha_A \cos \theta_j)^2 \quad (6.88)$$

$$\frac{D(6.85)}{\delta_{zB}} = \sum_{j=1}^Z K q (\delta_{xB} \cos \theta_j \cos \alpha_B + \delta_{xA} \sin \alpha_B + 0.5 P_{dB})^{\alpha-1} (\cos \alpha_B \cos \theta_j)^2 \quad (6.89)$$

High Speed Loads

At high speeds the effects of the centrifugal force on each roller may become significant. If the centrifugal force increases enough, the rolling elements may lose contact with the inner race. The lack of proper contact will produce a decrease in the load zone, increasing the load at each element, hence reducing the life of the bearing. The equations for high speed analysis are taken from reference [30]. The simpler case to consider is the case of cylindrical roller bearings under a radial external load.

Cylindrical Roller Bearings

The loads on a high speed CRB under an external radial load are illustrated on Figure 6.6. Besides the load at each roller/raceway contact, there is the centrifugal force acting outward on the centroid of the roller.

The equilibrium of forces is satisfied by

$$F_{o3} - F_{i3} - F_c = 0 \quad (6.90)$$

where

F_{o3} : Normal load at the roller/outer race contact
(N)

F_{i3} : Normal load at the roller/inner race contact
(N)

F_c : Centrifugal Force on the roller (N)

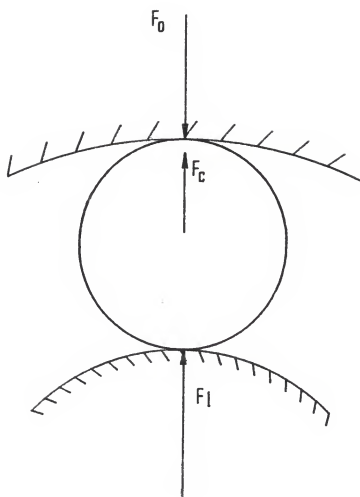


Figure 6.6 High Speed Loads on a Cylindrical Roller

Recalling equation (6.12), the load deflection relationship for a line contact is

$$F = k_x \delta^{1.11} \quad (N) \quad (6.12)$$

Substituting equation (6.12) into equation (6.90)

$$k_x \delta_{o_j}^{1.11} - k_x \delta_{i_j}^{1.11} - F_o = 0 \quad (6.91)$$

Dividing by k_x

$$\delta_{o_j}^{1.11} - \delta_{i_j}^{1.11} - F_o/k_x = 0 \quad (6.92)$$

The total radial δ_{θ_j} approach is the sum of the approach at each one of the two contacts.

$$\delta_{\theta_j} = \delta_{o_j} + \delta_{i_j} \quad (mm) \quad (6.93)$$

Expressing δ_{o_j} and substituting equation (6.93) into equation (6.92)

$$(\delta_{\theta_j} + \delta_{i_j})^{1.11} - \delta_{i_j}^{1.11} - F_o/k_x = 0 \quad (6.94)$$

Recalling equation (6.22), the deflection of a roller at the θ_j^{th} angular position is

$$\delta_{\theta_j} = \delta_x \cos \theta_j - 0.5 P_d \quad (mm) \quad (6.22)$$

Substituting equation (6.22) into equation (6.93), and this into equation (6.94) we obtain

$$(\delta_x \cos \theta_j - P_d/2 - \delta_{i_j})^{1.11} - \delta_{i_j}^{1.11} - F_o/k_x = 0 \quad (6.95)$$

The sum of all the radial loads at each inner race must equal the external applied radial load, assuming it is applied through the shaft. Recalling equation (6.25)

$$F_x = \sum_{j=1}^z k_x \delta_{\theta_j}^{1.11} \cos \theta_j \quad (N) \quad (6.25)$$

For z rolling elements, we can now solve for δ_{i_j} and δ_x since we have $z+1$ equations available. We obtain z equations

by expanding equation (6.94), letting $j=1,2,\dots,z$. Equation (6.24) is the $z+1$ equation. The equations can be expressed in terms of δ_{1j}

$$((\delta_x \cos \theta_j - P_d/2 - \delta_{1j})^{1.11} - F_c/k_x)^{9/10} = -\delta_{1j} \quad (6.96)$$

A vector of initial approximations of δ_{1j} is assumed with an initial value for δ_x . A new vector of δ_{1j} is computed using equation (6.96). To compute a new approximation for δ_x , equation (6.26) is used in conjunction with the Newton-Raphson method described above. First we determine the difference between the external applied radial load and the radial load computed using the deflections

$$D = F_x - \sum_{j=1}^z k_x \delta_{1j}^{1.11} \cos \theta_j \quad (6.97)$$

The partial derivative of (6.97) with respect to δ_{1j} is

$$DP = 1.11 \sum_{j=1}^z k_x \delta_{1j}^{0.11} \cos \theta_j \quad (6.98)$$

The new approximation of δ_x is given by

$$\delta_{x\text{new}} = \delta_{x\text{old}} + \frac{D}{DP} \quad (6.99)$$

By using this method, a solution for the high speed loading computations for cylindrical roller bearings was obtained. Once the computation of bearing loads at high speeds can be achieved, the effect of preload, radial loads, spindle speed and centrifugal loads on bearing deflections can be investigated. Computations were performed for a variety of operating conditions and the results indicate that at the operating speeds used in these tests the effect

of spindle speed on deflection is negligible, or none. The effect of radial load and preload are significant as would be expected. These results were expected based on the readings from the PROMESS sensor, which showed that the centrifugal forces on the bearings were quite low. For a double row cylindrical roller bearing, the measured centrifugal forces would not cause any significant deflection.

High Speed Tapered Bearings

The centrifugal loads induced by high speed operation of TRB are computed first. The gyroscopic moment effect will then be evaluated with its effects on the distribution of the load. These will then be combined with the effects produced by a combined loading situation. It is assumed that misalignment of the bearing does not occur.

Centrifugal Forces

The loads induced on a TRB by high speed operation are shown in Figure 6.7. The centrifugal load F_{cj} must be supported by the outer race contact and the flange contact. The sum of the axial and radial forces must equal 0.

$$F_{cj} - C_{ojradial} - C_{fjradial} = 0 \quad (6.100)$$

$$C_{fjaxial} - C_{ojaxial} = 0 \quad (6.101)$$

where

F_{cj} : Centrifugal force acting on the j^{th} roller
(N)

$C_{ojradial}$: Radial component of the outer race
reaction to the centrifugal force (N)

$C_{fjradial}$: Radial component of the flange reaction
to the centrifugal force (N)

$C_{fjaxial}$: Axial component of the flange reaction
to the centrifugal force (N)

$C_{ojaxial}$: Axial component of the outer race
reaction to the centrifugal force (N)

Rewriting equations (6.100) and (6.101) in terms of
 F_{oj} , C_{oj} and C_{fj}

$$F_{oj} - C_{oj}\cos\alpha_o - C_{fj}\cos\alpha_f = 0 \quad (6.102)$$

$$C_{fj}\sin\alpha_f - C_{oj}\sin\alpha_o = 0 \quad (6.103)$$

Solving in equation (6.103) for C_{fj} in terms of C_{oj}

$$C_{fj} = C_{oj} (\sin\alpha_o/\sin\alpha_f) \quad (6.104)$$

Substituting equation (6.104) into equation (6.102) and
solving for C_{oj} in terms of the centrifugal force

$$F_{oj} - C_{oj}\cos\alpha_o + C_{oj}(\sin\alpha_o/\sin\alpha_f)\cos\alpha_f = 0 \quad (6.105)$$

$$F_{oj} - C_{oj}[\cos\alpha_o + (\sin\alpha_o\cos\alpha_f)/\sin\alpha_f] = 0 \quad (6.106)$$

$$F_{oj} - C_{oj}[\cos\alpha_o\sin\alpha_f + \sin\alpha_o\cos\alpha_f]/\sin\alpha_f = 0 \quad (6.107)$$

$$C_{oj} = F_{oj} [\sin\alpha_f / (\sin(\alpha_f + \alpha_o))] \quad (6.108)$$

Substituting then equation (6.108) into equation
(6.104)

$$C_{fj} = F_{oj} [\sin\alpha_o / (\sin(\alpha_f + \alpha_o))] \quad (6.109)$$

When all the axial components of the centrifugal force
reaction on the outer race are added, they act as a thrust
load which tends to reduce the preload. The magnitude of
this induced thrust load is

$$F_{ac} = z C_o \sin \alpha_o \quad (N) \quad (6.110)$$

where

F_{ac} Centric axial load induced by the centrifugal forces acting on the rollers (N)

Gyroscopic Moment

The gyroscopic moment acting on each roller induces a redistribution of the loads on the cup race contact [37]. The magnitude of the gyroscopic moment is given by

$$M_{gj} = 8.37 \times 10^{-12} D^4 w_{mj} w_{Rj} \sin[0.5(\alpha_1 + \alpha_o)] \quad (6.111)$$

Assuming a linear load distribution along the length of the roller, the distributed load at the back of the roller is q_1 and at the front is q_2 . In terms of the centrifugal load C_o , and gyroscopic moment M_{gj} , q_1 and q_2 can then be determined by taking moments about the roller center of gravity. From the figure describing the geometry of the tapered bearing

$$C_o(A + x) + M_{gj} = C_x B \quad (6.112)$$

solving for x

$$x = \frac{(C_x B - M_{gj})}{C_o} - A \quad (6.113)$$

The center of gravity for a trapezoid load distribution is given by

$$x = \frac{L}{2} - \frac{L(q_1 + 2q_2)}{3(q_1 + q_2)} \quad (6.114)$$

and

$$(q_1 + q_2)L = 2C_o \quad (6.115)$$

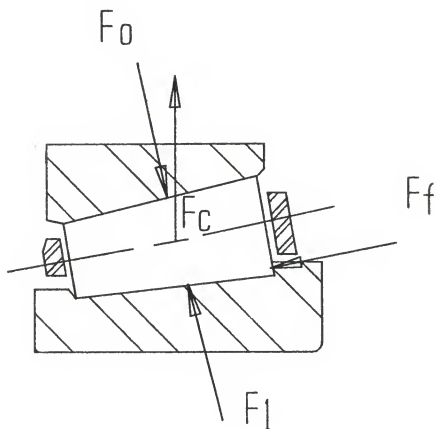


Figure 6.7 Loads on a High Speed Tapered Roller

Equating (6.113) and (6.114), substituting (6.115) into it and solving for q_2

$$q_2 = \frac{C_2}{L \left(\frac{1-6(C_2B-C_0A)}{C_0L} \right)} + \frac{6M_{GJ}}{L^2} \quad (6.116)$$

q_1 is given by

$$q_1 = \frac{C_2}{L \left(\frac{1+6(C_2B-C_0A)}{C_0L} \right)} - \frac{6M_{GJ}}{L^2} \quad (6.117)$$

Combined Loading

The load deflection equations for a TRB under combined loading and operating at high speeds are similar to those under static load. To the static load deflection equations we add the effect of the centrifugal force acting on the bearings, computed in the section on combined loading. The equilibrium equations for the high speed case will be

$$F_{rj} = F_{Gj} + k\delta_{rj}^{1.11} \quad (N) \quad (6.118)$$

$$F_{aj} = F_{ac} + k\delta_{aj}^{1.11} \quad (N) \quad (6.119)$$

These equations are then solved in the same manner as presented above. For the operating speeds at which the tests were performed, the computations show that the effect of spindle speed on bearing deflections can be neglected. Again, the effect of increased external loads and preload

play a more relevant role in the deflection of the bearings. The observed loading on the bearings, using the PROMESS sensor, confirms these results, since no change in bearing load was detected during the test of Configuration II.

CHAPTER VII

CONCLUSIONS

Large bore, 100 mm diameter, tapered roller bearings and double row cylindrical roller bearings were investigated for their High Speed, High Power capabilities. Two spindle configurations were tested; each based on a different bearing type, to investigate their performance in terms of maximum speed, operating temperatures, and power requirements. Based on the experimental results, conclusions and recommendations are presented regarding the suitability of these configurations for HSHP use.

Spindle Configurations

Cylindrical Roller Bearings

Configuration I is based on double row cylindrical roller bearings of the NN30 K type. It has a very common spindle-bearing arrangement in which the CRBs receive the radial load while a double row angular contact ball bearing (ACBB) sustains the axial load, Figure 3.2. Lubrication to the bearings is provided via holes in the outer ring of the bearings. The oil flows into the bearing cavity between the

two rows of elements and then out each row, Figure 3.6. This configuration operated at a maximum spindle speed of 8,000 rpm with very high operating temperatures, Figures 5.10 and 5.12. Twice the ACBB failed when attempts were made to drive the spindle to 10,000 rpm. The power losses for the configuration reached 20 kW for the maximum speed and oil flow rate used during the test, Figures 5.18, 5.19, and 5.21.

Tapered Roller Bearings

Configuration II is based on tapered roller bearings and it is equipped with a mechanism which maintains a constant preload during the operation of the spindle. The bearing at the drive side is a Hydra-Rib_{TM}, Figure 3.4. The bearing at the tool side is a tapered bearing provided with cone-rib lubrication which makes it suitable for high speed operations, Figure 3.5. This configuration is also provided with jet lubrication to the front of the bearings. Configuration II operated successfully to 9,000 rpm with an oil flow rate of 2.3 lpm. At this speed, the operating temperature was 67°C, Figures 5.11 and 5.13, and 15 kW in power losses, Figures 5.18, 5.20, and 5.22. The operating temperatures and power losses for Configuration II were always lower than those of Configuration I for the same spindle speed and oil flow rate.

Experimental Conclusions

Empirical Equations

Equations were derived using the experimental data to relate operating temperature and power losses to the spindle speed and the oil flow rate. The equations derived describe the relationship between: bearing temperature and spindle speed, (5.1), bearing temperature and oil flow rate, (5.2), mechanical power losses and spindle speed, (5.6), hydraulic power losses and spindle speed, (5.8), and hydraulic power losses and oil flow rate, (5.9). These equations match the experimental data very well and could be used to estimate the behavior of the configuration within the test range, Figures 5.10 to 5.13 and Figures 5.19 to 5.22.

The experimentally derived equations were combined to obtain general equations which estimate either the operating temperature or the hydraulic power losses in terms of both spindle speed and oil flow rate, equations (5.3), (5.4) and (5.10). The general equation relating spindle speed and oil flow rate to bearing operating temperature matches experimental data as good as the individual relationships, Figures 5.14 to 5.17 and Figures 5.23 to 5.26.

Based on the experimentally developed equations, it can be concluded that speed has a direct effect on bearing operating temperature. As the spindle speed increases, so does the operating temperature of the bearing. The effect of spindle speed on temperature was much stronger for Configuration I than for Configuration II. This is due to

the greater amount of rolling elements in Configuration I. The larger the number of rolling elements, the larger the frictional losses, and thus, the higher temperature.

Increasing the oil flow rate decreases the bearing's operating temperature. The larger the amount of lubricant circulated through the bearing the larger the cooling effect. Configuration I was more responsive to increases in oil flow rate than Configuration II. As it was described before, most of the oil forced through the cylindrical roller bearings is splashed away, increasing the power losses and having little cooling effect. On the other hand, tapered bearings have an efficient flow of oil through the cavity. Thus, increasing the oil flow rate has very little effect on the operating temperature.

Two parameters were defined to evaluate the performance of each spindle independently of the machine tool on which it was installed. These parameters are the mechanical power losses and the hydraulic power losses. The mechanical power losses refers to the power required to drive the spindle to the desired speed without oil circulation. The hydraulic power losses refers to the power required to drive the spindle through the circulating oil at the desired speed. The total power required to drive a spindle to the desired speed is the sum of the mechanical power losses, plus the hydraulic power losses, and plus the motor/transmission power losses. The last term, the motor/transmission power losses is independent of the spindle.

The effect of spindle speed on hydraulic power losses is much stronger than the effect of oil flow rate, as it can be concluded from Figures 5.19 to 5.22 and equations (5.8) to (5.11). For the same combination of spindle speed and oil flow rate, the power losses for Configuration I are higher than for Configuration II as it can be seen from equations (5.8) and (5.11). This is caused by the inefficient flow patterns in the cylindrical roller bearings and the larger amount of rolling elements in Configuration I. The higher power losses for Configuration I is the cause for the configuration's higher operating temperatures. The larger the power losses, the more heat is generated, and thus, the higher the operating temperatures.

It can be observed from Figures 5.10 to 5.13 that there is an oil flow rate after which any further increase in flow has little effect on bearing temperature. By comparing the increase in power losses, Figures 5.19 to 5.22, to the decrease in temperature for a given oil flow rate an optimum flow rate can be selected. For Configuration I, the optimum oil flow rate selected was 2.3 lpm, while for Configuration II the optimum flow rate was selected as 1.0 lpm, 0.5 lpm provided by the oil jets and 0.5 lpm provided through the cone-rib contact. These flow rates would provide the best combination of operating temperature and power losses. The maximum operating spindle speed recommended for each configuration based on operating temperatures and power losses are 6,000 rpm for Configuration I and 9,000 rpm for

Configuration II. At these spindle speeds, the power losses and the operating temperatures are acceptable for HSHP machining operations.

Bearing Preload

The PROMESS sensor indicated that the preload in Configuration I varied during the test. The change on preload was due to thermal expansions and to high speed centrifugal effects. The magnitude of the thermally induced preload as measured by the PROMESS sensor, 200 N, is negligible for the double row large diameter bearings used.

The effect of high speed centrifugal effects on bearing preload is of special consequence to the Radiax bearing. This was the bearing which failed twice during the test. As it was mentioned in Chapter V, the preload on the upper row was eliminated at high spindle speeds. Without preload the balls started to skid, which increased the heat generation rates. The lubrication system was unable to remove the excess heat induced by skidding and the bearing seized. Therefore, if Configuration I is to be used at speeds higher than 8,000 rpm, the mounting preload in the Radiax bearing must be increased.

The preloading mechanism used in Configuration II was effective throughout the test. At all times during the test, the PROMESS sensor indicated that the preload was kept constant. Therefore, the use of a constant preloading mechanism is recommended in HSHP spindles.

Recommendations

Design Modifications for Configuration I

If Configuration I is to be used above 6,000 rpm, the lubrication/cooling system must be modified. The modification must reduce the operating temperatures and decrease the hydraulic power losses. It is important to note that circulating oil lubrication provides an excess of lubricant to the bearing. Thus, reducing the amount of lubricant to improve the operating behavior does not affect the basic lubrication needs of the bearing. As it was shown earlier, increasing the oil flow rate produces both a decrease in the bearing's operating temperature and an increase in hydraulic power losses. An oil flow rate of 2.3 lpm was recommended above, since it would provide the best combination of operating temperatures and power losses.

Supplying oil directly to the elements would improve the cooling efficiency. Currently, all the oil is supplied through the outer race of the bearings, between the rows of elements. As the oil circulates through the bearing cavity, its temperature increases and it induces hydraulic power losses. If the oil is supplied directly from the source to the rolling elements, the oil reaching them would be much cooler than if it is first forced through the bearing. Since this oil would be splashed away from the elements, the hydraulic power losses would be reduced, decreasing the temperature even further.

To provide the element rows with lubrication the housing would have to be redesigned to include more oil input points. These additional oil inlet points could be used for oil jets directed to the element rows. Some of the locating rings currently used on the spindle would also have to be modified to allow oil to go through them and onto the rolling elements. These changes would improve the performance of the configuration by decreasing the power losses and the operating temperature. If the power losses decrease, then more power would be available for machining. The lower temperatures would improve the accuracy of the configuration and the life of the elements.

Design Modifications for Configuration II

The oil distribution system for Configuration II should be modified. The ring which supplies the lubricating oil to the Hydra-Rib_{TM} should be modified as to improve the oil flow into the bearing. Currently, it has eight lubrication holes. Since the oil is forced up the bearing, the pressure drop caused by the large number of holes reduces the oil flow into the bearing. If only three evenly spaced holes are provided the oil would reach the bearing with enough pressure to flow through. This would be a more efficient design.

Rings should be added to the spindle to splash away the excess oil from the bearings and into grooves in the housing. This modification would limit the amount of oil

reaching the lower seals. As currently designed, the excess oil from the top bearing plus all the oil from the lower bearing falls into the lower seal straining its capacity. The above two measures would lessen the amount of oil reaching the lower bearing, reducing drastically the amount of mist being produced.

To improve the survivability of the tapered bearings, the rib supporting the back of the bearing should be provided in the outer race. Currently, the rib is located at the back of the inner race. Due to the inherent oil flow patterns of the tapered bearings, the contact between the back of the bearing and this rib does not receive the necessary lubrication. Cone-rib lubrication was developed to solve this particular problem. Changing the rib to the outer race would use the inherent flow patterns to lubricate the cone-rib contact. Tapered bearing manufacturers have in stock a bearing with such a design. This bearing is not manufactured with the precision levels required for machine tools. Bearing manufacturers should provide a sample of these bearings to researchers to investigate their high speed performance.

Final Comment

Two spindle configurations based on different types of roller bearings were tested for High Speed, High Power performance. Although the target speeds were not met, their

performance guarantees the maximum use of the cutting tools available for the milling of steel and cast iron. Parameters were identified which can be used to compare the performance of a configuration independently of the machine tool in which they are installed. A set of empirical equations were derived to estimate the power losses at operating conditions different from those in the test. Finally, recommendations were presented for the design of spindle configurations based on roller bearings in terms of lubrication arrangements, operating speeds, and oil flow rates.

APPENDIX

COMPUTER PROGRAM LISTINGS

Radial Load Computation Program

```
10 REM RADIAL LOAD COMPUTATION PROGRAM
20 DIM F(20),DR(50),DTH(20)
30 PRINT "USE CONSISTENT UNITS":INPUT "EXTERNAL RADIAL FORCE
(N/LB)";FR
40 INPUT "DIAMETRAL CLEARANCE (mm/in)";PD
50 INPUT "ELEMENT'S LOAD-DEFLE. CONST (n/mm or lb/in)";KN
60 INPUT "NUMBER OF ELEMENTS";Z:INPUT "EXPONENT Q ";Q
70 INPUT "RESULTS ON SCREEN OR PRINTER (S/P)";X$
80 IF X$="S" THEN GOTO 120
90 LPRINT "RADIAL FORCE";FR:LPRINT "DIAMETRAL CLEARANCE ";PD
100 LPRINT "LOAD DEFLECTION CONSTANT";KN:LPRINT "NUMBER OF
BALLS";Z
110 LPRINT "EXPONENT Q";Q:GOTO 90
120 PRINT "RADIAL FORCE";FR:PRINT "DIAMETRAL CLEARANCE ";PD
130 PRINT "LOAD DEFLECTION CONSTANT";KN:PRINT "NUMBER OF
BALLS";Z
140 PRINT "EXPONENT Q";Q
150 J=-1:E=360/Z:DR(0)=.001
160 J=J+1:A=0:C=0
170 FOR I=0 TO Z-1:B= COS(3.1415926#*I*E/180)
180 DTH(I)= (DR(J)*B-.5*PD):IF DTH(I)<0 THEN DTH(I)=0
190 F(I)=KN*(DTH(I)^Q):A=(F(I)*B)+A
200 C=(Q*KN*(DTH(I))^(Q-1))*B^2+C:NEXT I
210 T=(FR-A)/C :IF T=0 THEN 260
220 DR(J+1)=DR(J)+T:V=ABS((DR(J+1)-DR(J))/DR(J+1))
230 IF V<.001 THEN 250:GOTO 160
240 GOTO 160
250 IF X$="S" THEN GOTO 320
260 LPRINT "RESULTS":LPRINT "          BALL          DEFLECTION
FORCE          "
270 FOR L=0 TO Z-1:M=L*E
280 LPRINT USING "#####.#####";M,DTH(L),F(L)
290 NEXT L
300 LPRINT "RADIAL DEFLECTION ",USING "###.#####";DR(J)
310 LPRINT "ITERATION ",USING "##.";J:GOTO 380
320 PRINT "RESULTS":PRINT "          BALL          DEFLECTION
FORCE          "
330 FOR L=0 TO Z-1:M=L*E
340 PRINT USING "#####.#####";M,DTH(L),F(L)
350 NEXT L
360 PRINT "RADIAL DEFLECTION ",USING "###.#####";DR(J)
370 PRINT "ITERATION ",USING "##.";J
380 END
```

Combined Load Computation Program

```

5 REM COMBINED LOAD COMPUTATION PROGRAM
10 DIM F(40),DR(30),DTH(40),DA(30)
20 INPUT "FR,FA,PD,KN,Z,Q";FR,FA,PD,KN,Z,Q
30 INPUT "CONTACT ANGLE";ALP
40 LPRINT "RADIAL FORCE";FR:LPRINT"AXIAL FORCE ";FA
50 LPRINT "DIAMETRAL CLEARANCE";PD
60 LPRINT"LOAD DEFLECTION CONSTANT";KN:LPRINT"NUMBER OF
BALLS";Z
70 LPRINT "EXPONENT Q";Q
80 LPRINT "CONTACT ANGLE ALPHA ";ALP
90 J=-1:G=COS(3.1412*ALP/180):H=SIN(3.1412*ALP/180)
100 E=360/Z:DR(0)=.001:DA(0)=.001
110 J=J+1
120 A=0:R=0:AD=0:RD=0
130 FOR I=0 TO Z-1
140 B= COS(3.1412*I*E/180):D=SIN(3.1412*I*E/180)
150 DTH(I)= (DA(J)*D+DR(J)*B*G-.5*PD)
160 IF DTH(I)<0 THEN DTH(I)=0
170 F(I)=KN*(DTH(I)^Q)
180 A=(F(I)*H)+A:AD=KN*Q*(DTH(I)^(Q-1))*H*D+AD
190 R=(F(I)*B*G)+R:RD=KN*Q*(DTH(I)^(Q-1))*(B*G)^2+RD
200 NEXT I
210 T=(FR-R)/RD:V=(FA-A)/AD
220 IF T=0 THEN 250
230 DA(J+1)=DA(J)+V:DR(J+1)=DR(J)+T
240 IF ABS((DR(J+1)-DR(J))/DR(J+1))>.001 THEN 110
250 LPRINT"RESULTS":LPRINT"          BALL          DEFLECTION
FORCE          "
260 FOR L=0 TO Z-1:M=L*E
270 LPRINT USING "#####.#####";M,DTH(L),F(L)
280 NEXT L
290 LPRINT"RADIAL DEFLECTION ",USING "#####";DR(J)
300 LPRINT"AXIAL DEFLECTION ",USING "#####";DA(J)
310 LPRINT "ITERATION ",USING "##.";J
320 STOP

```

Load Deflection Computation Program

```

10 REM RADIAL LOAD-DEFLECTION BEHAVIOR COMPUTATION PROGRAM
20 DIM F(20),DR(50),DTH(20),FF(50),DDR(50)
30 INPUT"INPUT BEARING SERIES";BEA$
40 PRINT "USE CONSISTENT UNITS"
50 INPUT "INITIAL EXTERNAL RADIAL FORCE (N/LB) AND
INCREMENT";FR,INC
60 INPUT "FINAL RADIAL FORCE (N/LB) ";FRF:FIN=INT((FRF-
FR)/INC)+1
70 INPUT"HOW MANY ROWS OF ELEMENTS";NROW
80 IF FIN>50 THEN GOTO 420:INPUT"HOW MANY ROWS OF
ELEMENTS";NROW
90 INPUT "DIAMETRAL CLEARANCE (mm/in)";PD
100 INPUT"ELEMENT'S LOAD-DEFLE. CONST (n/mm or lb/in)";KN
110 INPUT "NUMBER OF ELEMENTS/ROW";Z:INPUT "EXPONENT Q ";Q
120 INPUT "RESULTS ON SCREEN OR PRINTER (S/P)";X$
130 IF X$="S" THEN GOTO 180
140 LPRINT "BEARING TYPE ";BEA$
150 LPRINT "INITIAL RADIAL FORCE";FR:LPRINT"DIAMETRAL
CLEARANCE ";PD
160 LPRINT"LOAD DEFLECTION CONSTANT";KN:LPRINT"NUMBER OF
BALLS";Z
170 LPRINT "EXPONENT Q";Q:LPRINT "ROWS";NROW:J1=0:GOTO 230
180 PRINT "BEARING TYPE ";BEA$
190 PRINT "INITIAL RADIAL FORCE";FR:PRINT"DIAMETRAL
CLEARANCE ";PD
200 PRINT"LOAD DEFLECTION CONSTANT";KN:PRINT"NUMBER OF
BALLS";Z
210 PRINT "EXPONENT Q";Q:J1=0 :PRINT "NUMBER OF ROWS ";NROW
220 PRINT :PRINT :PRINT:PRINT:PRINT
230 J=-1:E=360/Z:DR(0)=.001:FR=FR/NROW
240 J=J+1:A=0:C=0
250 FOR I=0 TO Z-1:B= COS(3.1415926#*I*E/180)
260 DTH(I)= (DR(J)*B-.5*PD):IF DTH(I)<0 THEN DTH(I)=0
270 F(I)=KN*(DTH(I)^Q):A=(F(I)*B)+A
280 C=(Q*KN*(DTH(I)^(Q-1))*B^2+C:NEXT I
290 T=(FR-A)/C :IF T=0 THEN 340
300 DR(J+1)=DR(J)+T
310 V=ABS((DR(J+1)-DR(J))/DR(J+1))
320 IF V<.001 THEN 340
330 GOTO 240
340 FF(J1)=FR:FR=FR+INC:DDR(J1)=DR(J)
350 J1=J1+1:IF J1<FIN THEN 230
360 IF X$="S" THEN GOTO 390
370 FOR L=0 TO FIN-1:LPRINT USING"#####.#####";FF(L),DDR(L)
380 NEXT L:GOTO 410
390 FOR L=0 TO FIN-1:PRINT USING"#####.#####";FF(L),DDR(L)
400 NEXT L
410 END
420 PRINT "CHANGE THE DIMENSION OF FF AND DDR":STOP

```

```

      High Speed Cylindrical Roller Bearing Program
10 REM CYLINDRICAL HIGH SPEED BEARING
20 LPRINT CHR$(12)
30 DIM DR(21),D(25,21),F(21)
40 INPUT "FR,PD,KN,Z,Q,N,FI";FR,PD,KN,Z,Q,N,FI
50 LPRINT "RADIAL FORCE (LB) ";FR:LPRINT "DIAMETRAL
CLEARANCE ";PD
60 LPRINT "STIFFNESS (LB/IN) ";KN:LPRINT "NUMBER OF
ELEMENTS ";Z
70 LPRINT "EXPONENT ";Q:LPRINT "SPINDLE SPEED (RPM) ";N
80 E=360/Z:PI=3.1412:DR(1)=.0002:FC=(N^2)*1.772E-14
90 FOR I3=1 TO Z:D(I3,1)=.0001:NEXT I3
100 FOR J=1 TO FI
110 C=0:FF=0
120 FOR I=1 TO Z
130 B= COS((I-1)*E*PI/180)
140 AA=(DR(J)*B-PD/2-D(I,J)):IF AA<0 THEN AA=0
150 A= (AA)^Q-D(I,J)^Q-FC/KN
160 AAD= (DR(J)*B-PD/2-D(I,J)):IF AAD<0 THEN AAD=0
170 AD= Q*(AAD)^(Q-1)+Q*D(I,J)^(Q-1)
180 D(I,J+1)=D(I,J)+A/AD:IF D(I,J+1)<0 THEN
D(I,J+1)=ABS(D(I,J+1))
190 F(I)=D(I,J+1)^Q*KN*B:FF=FF+F(I):C=Q*KN*B^2*D(I,J+1)^(Q-
1)+C
200 NEXT I
210 AR=(FR-FF)/C
220 DR(J+1)=DR(J)+AR
230 NEXT J
240 LPRINT USING "RADIAL DEFLECTION = #####.#####";DR(J)
250 LPRINT "ANGULAR POSITION DEFLECTION FORCE
":FOR I1=1 TO Z
260 X=(I1-1)*E
270 LPRINT USING " #####.#####";X,D(I1,J), F(I1)
280 NEXT I1
290 STOP

```

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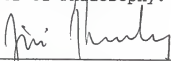
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BIOGRAPHICAL SKETCH

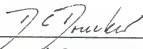
The author was born December 4, 1959, in Mayagüez, Puerto Rico. He went to college at the University of Puerto Rico, at Mayagüez, obtaining a B.S.M.E. in June 1982, and accepted a faculty position with the UPR. He began graduate study at Clemson University in Clemson, South Carolina, where he completed his M.S.M.E. in December 1983. He married his beautiful and enchanting wife Laura M. on January 2, 1984. After completing his Ph.D. from the University of Florida, he expects to return to his position at UPR.

I certify that I have read this study and that in my opinion it conforms to acceptable standards of scholarly presentation and is fully adequate, in scope and quality, as a dissertation for the degree of Doctor of Philosophy.



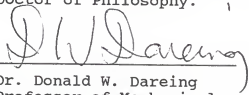
Dr. Jiri Tlustý, Chairman
Graduate Research Professor
of Mechanical Engineering

I certify that I have read this study and that in my opinion it conforms to acceptable standards of scholarly presentation and is fully adequate, in scope and quality, as a dissertation for the degree of Doctor of Philosophy.



Dr. Daniel C. Drucker
Graduate Research Professor
of Aerospace Engineering,
Mechanics, and Engineering
Science

I certify that I have read this study and that in my opinion it conforms to acceptable standards of scholarly presentation and is fully adequate, in scope and quality, as a dissertation for the degree of Doctor of Philosophy.



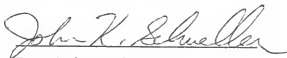
Dr. Donald W. Dareing
Professor of Mechanical
Engineering

I certify that I have read this study and that in my opinion it conforms to acceptable standards of scholarly presentation and is fully adequate, in scope and quality, as a dissertation for the degree of Doctor of Philosophy.



Dr. Charles L. Proctor II
Associate Professor of
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I certify that I have read this study and that in my opinion it conforms to acceptable standards of scholarly presentation and is fully adequate, in scope and quality, as a dissertation for the degree of Doctor of Philosophy.



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This dissertation was submitted to the Graduate Faculty of the College of Engineering and to the Graduate School and was accepted as partial fulfillment of the requirements for the degree of Doctor of Philosophy.

May 1989



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